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THE U. S. AIRPLANE CARRIERS *LEXINGTON* AND *SARATOGA*. A STUDY IN SCREW PROPULSION.

BY REAR ADMIRAL C. W. DYSON, U. S. N., RET., MEMBER.

INTRODUCTION.

The United States Airplane Carriers *Lexington* and *Saratoga* were originally designed as Battle Cruisers, together with four sister ships, but by the Treaty for the Limitation of Armaments adopted in 1922, the four sister ships were condemned to the scrap heap and permission was granted to convert these two ships to airplane carriers.

In the transformation of these vessels from battle cruisers to airplane carriers, the major part of the necessary changes was in the hull. The underwater body lines were modified by the addition of bulges for protection against torpedo attack.

PRELIMINARY ACCEPTANCE TRIALS

TABLE I

"SARATOGA" AND "LEXINGTON"

SECOND PRELIMINARY ACCEPTANCE TRIALS.

Speed		10	12	14	16	18	20	22	24	26
S.H.P.	SARATOGA	3900	6200	10000	15500	22500	31500	42100	53500	66800
	LEXINGTON	3900	6600	10200	15300	21500	29900	40500	52500	65500
Revs.	SARATOGA	85.5	102.2	119	136.5	153.6	170.5	188	204.6	221.5
	LEXINGTON	85	102	118.5	136.1	153	170	187	204	221

Speed		28	30	31	32	33	34	34.59	34.64	
S.H.P.	SARATOGA	81500	100000	113000	130500	156000	189000	210000	212300	
	LEXINGTON	79500	98500	111000	127000	149000	180000	209700		
Revs.	SARATOGA	238	255.5	264.5	278	293	311	324	325	
	LEXINGTON	236	257	266.5	277	291.5	307	320		

First Trials of LEXINGTON and SARATOGA

Speed		10	12	14	16	18	20	22	24	26	28	30
S.H.P.	SARATOGA	3800			15750		30600		52800		81200	101500
	LEXINGTON	4000			15250		31300		54400		83500	101000
Revs.	SARATOGA	85.4			138		173.3		207.7		240	259.5
	LEXINGTON	87			138.8		175.2		210.2		241.9	259.1
Mean S.H.P.	1st	3750			15500		30950		53450		82350	101250
	2d	3900			15350		30650		53000		80500	99250
Mean Revs.	1st	86.2			138.4		174.25		208.95		240.95	259.3
	2d	85.25			136.5		170.25		204.3		238	256.1

Speed		31	32	33
S.H.P.	SARATOGA		132500	156575
	LEXINGTON		130000	153600
Revs.	SARATOGA		282.5	295.6
	LEXINGTON		281.6	297
Mean S.H.P.	1st		131250	155088
	2d		128750	152500
Mean Revs.	1st		282.05	296.3
	2d		277.5	292.25

The only changes required in the machinery installations were the moving of the smoke pipes from the midship line out to the starboard side of the vessels and enclosing all the pipes in one outer casing, and the designing of new propellers made necessary by the change in the underwater lines of the hulls.

The two vessels were completed late in the Fall of 1927 and were turned over to the Navy. After several months of training, the *Lexington* was given her first acceptance trials in the early Spring of 1928, followed by those of the *Saratoga* in June.

FIRST TRIALS.

Shortly before the first trials of the *Lexington* an inspection of the turbine blading disclosed that the rotor bladings of the 12th and 13th stages were showing evidences of weakness. A similar inspection of the turbines of the *Saratoga* disclosed the same fault.

The rotor blading of these two stages in the turbines of both vessels were removed and instructions were given that on the first trials the turbines should only be driven up to about 150,000 shaft horsepower, which instructions were adhered to.

The weather during the first trials of the *Lexington* was inferior while during those of the *Saratoga* it was good. The effect of the difference in weather conditions is shown by the tabulated results given in Table 1.

Upon completion of these trials the vessels proceeded to the Puget Sound Navy Yard where blading of greater strength than the original was installed in the 12th and 13th stages of the turbines.

As soon as this work was finished, the vessels were dry-docked at Hunter's Point, San Francisco, and the bottoms thoroughly cleaned and painted. This precaution had also been taken in preparing for the first trials.

SECOND TRIALS.

After the vessels were undocked the vessels were taken to the carefully measured deep water course in Santa Barbara Channel, the same course over which the first trials had been conducted. The *Lexington's* trials started on November 12 and ended on

November 16, 1928, while those of the *Saratoga* extended from December 12 to December 17. The weather conditions during these trials were good for both vessels.

The data as to speeds, power and revolutions are given in Table 1. This table also gives a comparison of the mean performances of the two vessels for the two sets of trials throughout the range of speeds covered by the first ones. It will be seen that there was a considerable improvement on the second trials, the excess power shown by the mean data of power for the first trials being due to the inferior weather condition effect upon the performance of the *Lexington.*

The trials of these ships are of the greatest interest to those interested in the propeller problem, not only to those who base their studies upon the performances of actual ships and propellers but also to those who use the Model Tank with its self-propelled models as the basis from which to draw their conclusions.

These trials are not only of interest on account of the great size and importance of the vessels, but on account of the enormous range in shaft horsepower, the great range in speed, the high tip speeds for such large propellers and the very high effective thrusts to which the propellers were subjected without the production of any evidences of cavitation.

On account of this interest the performances of the second trials will now be studied in detail.

HULL CHARACTERISTICS AFFECTING THE PROPELLERS.

Length on Load Water Line	= L.L.W.L.	= 850′
Beam	= B	= 106′
Draft	= H	= 27.′88
Maximum height due to possible squat of stream line body	= H′	= 32′
Beam ÷ Length on Load Water Line	= B ÷ L.L.W.L.	= .1247
Nominal Block Coefficient	= N.B.C.	= .545
Coefficient of Midship Section	= Coef. M.S.	= .992
	H′ ÷ B	= .3
	H ÷ B	= .263
	Type	= 1
Slip Block Coefficient	= S.B.C.	= .415

2 × Length of after body	= 2 × L.A.B.	= 850′
For Maximum Squat	(*a*) 2 L.A.B. ÷ H′	= 26.56
For no Squat	(*b*) 2 L.A.B. ÷ H	= 30.5
Basic apparent slip for (*a*)	= S	= .1625
Basic apparent slip for (*b*)	= S	= .138
Power Loss Factor Block Coefficient	= K.B.C.	= .495
Mean Tip Clearance	= M.T.C.	= Large
Power Loss Factor	= K	= 1

These hulls may be classed as gigantic light cruiser or still more gigantic destroyer hulls and as such may be expected to exhibit in their performances all of the peculiarities of these lesser ships. All of these vessels, according to the past practice of our designers have < shaped lines at the load water line aft and squat very considerably as the speed increases; with the increasing squat the wake gradually increases and the basic apparent slip decreases until a limiting speed v, dependent upon the value $\frac{v}{\sqrt{L.\ L.\ W.\ L.}}$, is attained. Upon passing this speed the wake gradually decreases and the basic apparent slip increases until, should the necessary speed to produce the maximum amount of squat be realized, the basic apparent slip will have attained the figure given by (*a*) in the above data, while if no squat is realized the basic apparent slip will be as given by (*b*). The propeller designer having no data from which to determine the amount of squat at full speed or at any other, for that matter, must be satisfied to state that the basic apparent slip at full speed will be somewhere between the two slips given by (*a*) and (*b*).

THE SCREW PROPELLERS.

These are four in number for each vessel. They are of manganese bronze, three-bladed, of uniform pitch and have the driving faces machined to the designed pitch, while the backs of the blades are made to template in order to insure proper blade sections and thickness. The projected area form is standard oval.

MEAN BASIC CONDITIONS OF THE PROPELLERS.

The conditions of wake for the inboard and the outboard propellers differing, in place of determining the basic condition for

each propeller according to its location, the mean condition for the two propellers is determined; thus

Diameter of propeller in feet = D = 14′ — 10⅝″

Pitch of propeller in feet = P = 13′ — 3″

Projected area ratio outside. $\frac{2D}{D.A.} = \frac{P.A.}{D.A.} = .65$

Basic Tip Speed in feet per minute = T.S. = 15196

$$= 38146 \left(\frac{P.A.}{D.A.}\right)^{.8365 + 2\left(\frac{P.A.}{D.A.}\right)}$$

This is the approximate tip speed at which this projected area ratio will enter cavitation due to excess tip speeds.

Assumed Basic Slips = S = .10 ..... .15 ..... .20

As it is desired in the investigation of performances to determine the variations of wake, as shown by the variations in the values of S, these assumed values are used instead of the values given by (*a*) and (*b*).

$1 - S = .90$ ..... $.85$ ..... $.80$

Corresponding Basic Speeds = V = 38.39 ..... 36.26 ..... 34.12

$$= \frac{T.S. \times P \times (1 - S)}{101.33 \times \pi \times D}.$$

Basic Shaft Trust in pounds per square inch of Disc Area =

$$S.T._D = 12.55 = 26.267 \times \left(\frac{P.A}{D.A.}\right)^{1.7}.$$

Basic Shaft horsepower = S.H.P. = 164120 (Total for 4 propellers)

$$= \frac{T.S. \times P \times D^2 \times S.T._D}{\pi \times D \times 291.8}$$

$$= \frac{P \times R \times D^2 \times S.T._D}{291.8}$$

Empirical Basic Propulsive Coefficient = $P.C._s = .571$

Basic effective horsepower = E.H.P. = S.H.P. × $P.C._s$ = 93715.

THE STUDY OF THE SECOND TRIALS.

Calling any value $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ the net load fraction where e.h.p. is the model tank effective horsepower for any speed v, values of the net load fraction are given in Column 1, Table 2, while the corresponding values of v and e.h.p., as taken from the model tank curve are shown in Columns 3 and 2. In Column 9 are shown the mean shaft horsepower for these same values of v as determined by the trials.

The plotted values of $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ of e.h.p. and of S.H.P._d are shown on Figure 1, plotted on v as abscissae. These curves are given the same numbers as the column numbers for the same data in Table 1; thus curve of $\frac{\text{e.h.p.}}{\text{E.H.P.}}$ is marked 1; of v, 3; of e.h.p., 2; and of S.H.P._d, 9.

The curve of actual revolutions as determined from the trials is shown on Figure 1, marked 20.

TO DETERMINE BASIC SLIPS FROM ACTUAL TRIAL RESULTS.

From the trials of many vessels, where the propellers were not in cavitation, the equation for apparent slip is

$$s = S \times 10^{Z_s} \times 10^{Z_g} \text{ where}$$

$$Z_s = 2.869 \text{ Log } \frac{V}{v} - .0788 \text{ and}$$

$$Z_g = \text{Log S.H.P.}_d - \text{Log S.H.P.},$$

$$\text{and Revolutions} = R_d = \frac{v \times 101.33}{P \times (1-s)}.$$

Assuming values of S of .10, .15 and .20, with corresponding values of V, curves of revolutions erected on values of S as abscissae are obtained. From these curves the values of S corresponding to the actual revolutions on trial, as shown by curve 20, Figure 1, are obtained. These values of S are given in Column 5, Table 2, while the corresponding values of V are given in Column 6.

These values of S are shown plotted as curve 5 on Figure 1, and the faired curve of S is shown as dotted curve 17.

The values of $v \div V$, Column 3 of Table 1, divided by Column 6, are given in Column 7.

TO FIND w AND A AND EQUATION FOR POWER.

In developing the formulas showing the relationship between actual indicated or shaft horsepower and basic indicated or shaft horsepower, it was assumed that it was a function of the ratios between actual and basic speeds and also between actual gross and basis effective horsepowers. That is

$$\left.\frac{\text{I.H.P.}_d}{\text{S.H.P.}_d}\right\} = \left\{\frac{\text{I.H.P.}}{\text{S.H.P.}}\right\} \times \left(\frac{v}{V}\right)^w = \left\{\frac{\text{I.H.P.}}{\text{S.H.P.}}\right\} \times \left(\frac{\text{e. h. p.}_g}{\text{E.H.P.}}\right)^A$$

or

$$w = \frac{\text{Log H. P.}_d - \text{Log H. P.}}{\text{Log } v - \text{Log } V} = \frac{A\,(\text{Log e.h.p.}_g - \text{Log E.H.P.})}{\text{Log } v - \text{Log } V}$$

In selecting performances from which to determine the values of w, care was taken to use such vessels as had a fine after body form, with a power loss-factor, K, of unity. This causes the net effective and the gross effective horsepowers to be equal.

Solving the performances for different values of $\text{e.h.p.}_g \div \text{E.H.P.}$, the values of w were plotted as curves shown on Figure 2, curves of w erected on $v \div V$ as abscissae, being shown for values of $\text{e.h.p.}_g \div \text{E.H.P.}$ ranging from .025 to 2. When the actual engine power becomes equal to the basic, e.h.p._g equals E.H.P. and w becomes zero. When e.h.p._g becomes greater than E.H.P., w changes sign and becomes negative. These curves only hold true for values of $v \div V$ equal to unity or lower and provided the propellers are not in cavitation due to excess thrusts.

Taking these values of w with the corresponding values of $v \div V$ and $\text{e.h.p.}_g \div \text{E.H.P.}$ and using them in the equation

$$A \text{ Log } (\text{e.h.p.}_g \div \text{E.H.P.}) = w \text{ Log } (v \div V),$$

$$\text{and } A = \frac{w \text{ Log } (v \div V)}{\text{Log } (\text{e.h.p.}_g \div \text{E.H.P.})},$$

A was found to have a constant value of 1.0414 for all values of e.h.p.$_g$ ÷ E.H.P. The value of A Log (e.h.p.$_g$ ÷ E.H.P.) = Log (H.P.$_d$ ÷ H.P.) is called Z, so that

$$Z = 1.0414 \text{ Log } (\text{e.h.p.}_g \div \text{E.H.P.}) \text{ and}$$

$$Z = \text{Log } (\text{H.P.}_d \div \text{H.P.}).$$

Z for e.h.p., the net effective horsepower is marked as Z_p, and as Z_g for the gross effective power.

The values of Z for the actual performances are given in Column 10, Table 2; those of A = Z ÷ Log (e.h.p. of Column 2 ÷ E.H.P.) in Column 11, while the values of w = Z ÷ Log (v ÷ V) are given in Column 12.

TO FIND THE ACTUAL GROSS LOAD FRACTIONS AND THE ACTUAL GROSS EFFECTIVE HORSEPOWERS DELIVERED AT SPEEDS V.

Column 10 of Table 2 shows the values Z_g equal to Log S.H.P. — Log S.H.P.$_d$. From these values of Z_g, the actual values of e.h.p.$_g$ ÷ E.H.P. can be obtained, as

$$\text{Log } (\text{E.H.P.} \div \text{e.h.p.}_g) = \frac{Z_g}{1.0414}.$$

The resultant values of e.h.p.$_g$ ÷ E.H.P. are given in Column 13, and those for $\text{e h.p.}_g = \text{E.H.P.} \times \frac{\text{e.h.p.}_g}{\text{E.H.P.}}$ are given in Column 14.

The curve of $\frac{\text{e.h.p.}_g}{\text{E.H.P.}}$ is shown dotted and marked 13 on Figure 2, while that for e.h.p.$_g$ is also shown dotted and marked 14.

TO ESTIMATE THE PROBABLE CURVE OF SHAFT HORSEPOWER FROM THE MODEL TANK CURVE OF EFFECTIVE HORSEPOWER.

When the designer of the propellers for a new vessel comes to take up the task of estimating the curve of shaft horsepower for a range of speeds, the only information as to resistance that he has available is the curve of effective horsepowers supplied him by the Model Tank.

TABLE 2

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22
$\frac{e.h.p.}{S.H.P.}$	e.h.p.	v	v_g	s	V	v + V	$v_g \div V$	S.H.P.$_d$	S	A	w	$\frac{e.h.p._c}{S.H.P._c}$	e.h.p.$_c$	S_g	S.H.P.$_{d_g}$	S	w_g	Est.R_d	Act.R_d	E.T.	$1 - \frac{v}{V_g}$
.085	2343	10.1	9.3	.112	37.88	.2667	.2446	3900	-1.6241	1.014	2.829	.02786	2584	-1.5694	3622	.109	2.731	79.93	79.5	1.26	-.108
.08	4486	12.2	12.8	.110	37.94	.3232	.3290	6600	-1.3956	1.078	2.845	.04569	4282	-1.368	7197	.110	2.822	108.9	106	1.874	-.085
.078	7029	14.6	14.4	.11	37.94	.3863	.380	11400	-1.1585	1.05	2.804	.07722	7236	-1.1715	11067	.1105	2.780	122.2	122.3	2.44	+.051
.1	9372	16.8	16.7	.107	38.09	.4253	.413	15800	-1.0165	1.017	2.736	.1067	9903	-1.0414	14930	.109	2.711	134.1	133.7	2.984	+.059
.15	14087	18.5	18.2	.107	38.09	.4687	.4768	23900	-.8586	1.016	2.668	.1572	14734	- .888	22760	.1045	2.668	154.7	155	3.838	+.05
.2	18745	20.5	20	.106	38.17	.537	.5225	33200	-.6940	.9929	2.57	.2156	20208	-.7279	30709	.102	2.582	170.1	170.3	4.686	+.081
.25	23429	22.1	21.8	.10	38.39	.5737	.5437	42800	-.5688	.978	2.485	.2720	25492	-.623	39099	.101	2.508	183.9	184.2	5.422	+.087
.3	28115	23.4	22.9	.10	38.39	.6096	.5965	49600	-.5197	.9939	2.416	.3169	29702	-.6445	46845	.099	2.425	194.9	195	6.137	+.089
.35	32800	24.7	24.39	.10	38.39	.6435	.6361	57700	-.454	.9967	2.371	.3645	34346	-.67	55611	.099	2.384	207.3	207.6	6.722	+.086
.4	37486	25.8	25.58	.10	38.39	.6721	.6661	64750	-.404	1.013	2.341	.4084	38566	-.6144	63206	.099	2.348	217.2	217.2	7.325	+.071
.45	42172	26.9	26.82	.10	38.39	.7006	.6992	72200	-.367	1.086	2.311	.4846	42596	-.56	71641	.10	2.316	227.4	228.7	7.868	+.008
.5	46858	27.9	27.9	.104	38.42	.7201	.7289	79600	-.3143	1.044	2.30	.4992	44779	-.3136	79737	.102	2.283	236.8	237.2	8.398	-.0019
.55	51543	28.85	28.9	.104	38.22	.7541	.7568	87500	-.2752	1.052	2.229	.5467	51230	-.27	88137	.104	2.231	245.7	246.2	8.915	-.0072
.6	56229	29.7	29.71	.109	38	.7815	.7806	95250	-.2318	1.045	2.165	.5990	56136	-.231	96418	.107	2.147	253.4	254	9.461	-.0017
.65	60915	30.3	30.36	.109	38	.7974	.7999	104000	-.2085	1.069	2.057	.6453	60472	-.195	104750	.109	2.006	260	260	10.05	-.0072
.7	65601	30.9	31.1	.113	37.81	.8173	.8217	110250	-.1788	1.116	1.972	.6825	63989	-.1618	113205	.112	1.891	266.9	267.6	10.54	-.0361
.75	70286	31.46	31.69	.110	37.94	.8362	.8396	118600	-.1411	1.132	1.842	.732	68605	-.127	122506	.1145	1.673	273.8	273.8	11.09	-.0319
.8	74972	31.98	32.05	.115	37.75	.8474	.8506	127750	-.1097	1.129	1.525	.7847	73540	-.1009	130096	.116	1.456	278.1	278	11.69	-.0199
.85	79658	32.4	32.5	.118	37.59	.8619	.864	137000	-.0784	1.111	1.215	.8408	78793	-.073	138735	.1195	1.168	282.8	284	12.25	-.0134
.9	84344	32.75	32.8	.1275	37.19	.8807	.879	145000	-.0538	1.175	.9745	.8879	83207	-.0477	149080	.1248	.8613	286.9	289	12.86	-.0139
.95	89029	33.1	33.15	.13	37.08	.8928	.8954	154500	-.0262	1.177	.5316	.9437	88434	-.0225	159906	.13	.4556	295	294	13.43	-.0115
1.00	93715	33.4	33.39	.135	36.87	.9101	.9046	163000	+.00832	--	-.00008	1.006	94198	0	164120	.134	0	297.4	298.1	14.03	+.006
1.05	98397	33.75	33.7	.15	36.58	.9316	.928	174000	+.0854	1.198	-.0006	1.069	99127	+.0821	172890	.148	-.00072	306.9	306	14.6	+.006
1.10	103079	34	33.95	.15	36.23	.9385	.9371	184000	+.0497	1.2	-.00157	1.116	104890	+.0431	181245	.15	-.00422	309.7	308.8	15.16	+.015
1.15	107760	34.18	34.2	.145	36.44	.9279	.9418	191500	+.0670	1.104	-.00187	1.16	106486	+.0682	192025	.148	-.00179	314	313.8	15.76	-.0067
1.2	112488	34.44	34.37	.145	36.44	.9469	.9442	200000	+.0859	1.085	-.00204	1.209	113510	+.0825	198456	.146	-.00206	317.4	317	16.34	+.088
1.25	117140	34.6	34.61	.142	36.87	.9461	.9464	210000	+.1071	1.105	-.00255	1.267	118784	+.1009	211865	.142	-.00242	320.4	321	16.92	-.009
1.30	121830	34.71	34.715	.14	36.66	.9469	.9471	216000	+.1173	1.034	-.00298	1.333	124930	+.1186	215655	.14	-.00281	323.1	324	17.51	-.005

Average difference ±.01164

In the case of the *Saratoga* and *Lexington,* this curve is shown on Figure 1, marked 2, and the data as to speed and effective horsepower are given in Columns 3 and 2, Table 2.

Using the load fractions as given in Column 1, Table 2, from the equations

$$Z = 1.0414 \text{ Log } \frac{\text{E.H.P.}}{\text{e.h.p.}_s}$$

for load fractions less than unity, and

$$Z = 1.0414 \text{ Log } \frac{\text{e.h.p.}_s}{\text{E.H.P.}}$$

for those greater than unity, the values of Z shown in Column 15, Table 2, are obtained. Then from

$$\text{S.H.P.}_d = \text{S.H.P.} \div 10^z; \frac{\text{e.h.p.}_s}{\text{E.H.P.}} < 1,$$

$$\text{S.H.P.}_d = \text{S.H.P.} \times 10^z; \frac{\text{e.h.p.}_s}{\text{E.H.P.}} > 1,$$

the values of S.H.P._d, shown in Column 16, Table 2, result.

These values of S.H.P._d are plotted on Figure 1, as curve 16, the abscissae being the Model Tank values of v given in Column 3, Table 2.

COMPARISON OF ACTUAL AND ESTIMATED SHAFT HORSEPOWER CURVES.

An examination of curves 9 and 16, Figure 1, discloses that from 12 knots speed to 27.5 knots, the actual shaft horsepowers slowly increase from equality with the estimated up to 22 knots where the percentage increase is greatest. From 22 knots to 27.5 knots the percentage difference gradually decreases until at 27.5 knots it has entirely disappeared.

The slight differences between the actual and the estimated shaft horsepowers from 27.5 knots to the maximum speed of 34.64 knots can be neglected as due to the smallness of the scale for power, the steepness of the power curves in this range and the thickness of the curve lines, errors which would produce these

small differences can readily be made. The percentage differences between the actual and the estimated shaft horsepower, that is,

$$1 - \frac{\text{Actual Shaft Horsepower}}{\text{Estimated Shaft Horsepower}},$$

are shown in Column 22, Table 2.

The question which naturally suggests itself concerning these differences is "are they due to an error inherent in the method of making the estimate?"

The answer to this question is "decidedly, No." In support of this answer, while the writer has had no opportunity to personally see them, he has been informed by persons in positions of authority at the Model Tank that the self-propelled model shaft horsepower curve for these vessels follows curve 16, Figure 2, very closely but is a little below it.

The answer to the first question being in the negative, it may be again asked, "What was the reason for the differences?"

The difference between curves 9 and 16 from 12 knots to 27.5 knots is such as would be produced by wind and sea or current, or, possibly all three combined, striking diagonally across the trial course and causing the vessel to drift slightly to leaward. The amount of this drift would depend upon the drifting force and upon the length of time on the course. This drift would require the use of a helm angle to keep the ship's head pointing in the direction of the end buoy of the course, if one were used.

As the speed increased from 12 to 22 knots, the increase in speed, from curve 9, causes an increase in percentage of resistance although the rudder angle has been slowly decreasing. From 22 to 24 knots the increase in resistance due to the rudder angle remains constant although this angle is slowly decreasing, but the percentage resistance is decreasing. After passing 24 knots, both the actual amount of and the percentage increase in resistance due to the rudder angle decrease rapidly as the speed of the ship increases and the time on the course decreases, until at 27.5 knots the time on the course is so short and the possible drift in that time so small, that the use of the rudder is no longer required. It is therefore kept amidships and so remains throughout all the higher speeds.

If the above explanation is to be accepted it should be supported by evidence showing that with increase in drifting force there is a corresponding increase in shaft horsepower required and that the condition with rudder amidships is not reached until a still higher speed than 27.5 knots has been attained.

This evidence is provided by the first trials of the *Lexington*, when with inferior weather, heavier winds and seas than experienced during the second trial, the entire shaft horsepower curve, up to 33 knots, was raised and amidship rudder apparently not used until a speed of 30 knots had been reached.

The mean arithmetical difference between the mean of the shaft horsepowers for the two ships and the estimated, only amounts to 2.82 per cent while the algebraic mean is only 1.164 per cent. For the best performance these figures reduce to 2.26 per cent and .608 per cent.

CORRECTION OF SPEEDS FOR S.H.P.$_d$ OF COLUMN 16, TABLE 2. ESTIMATE OF REVOLUTIONS FOR THESE SPEEDS.

Turning again to Figure 1, curve 9, it will be seen that for the S.H.P.$_d$ values given in Column 16, Table 2, the corresponding speeds are those given in Column 4. From curve 17 of the same Figure it is seen that the basic slips, S, for these new speeds are those given in Column 17, Table 2.

New values of V and of $v \div V$ for these values of S were computed, the new values of $v \div V$ being tabulated in Column 8, Table 2.

Using these new values of $v \div V$ and the power factor formula

$$Z = 1.0414 \text{ Log } \left(\frac{\text{e.h.p.}_g}{\text{E.H.P.}}\right), \text{ where all values of Z}$$

are additive, or

$$Z = 1.0414 \text{ Log } \left(\frac{\text{E.H.P.}}{\text{e.h.p.}_g}\right) \text{ for values of } \frac{\text{e.h.p.}_g}{\text{E.H.P.}}$$

less than unity, where values of Z are subtractive, new values of w were obtained. These are shown in Column 18, Table 2, and are also shown plotted, marked ⊙, on Figure 2, the original values of w, given in Column 12, also being shown plotted for $v \div V$, Column 7, and marked X.

In making the estimate of revolutions from approximate actual conditions of wake as shown by curve 17, Figure 1, use the equation

Apparent slip $= s = S \times 10^{Z_s} \times 10^{Z_g}$, where the values of S are taken from Column 17, Table 2, and where

$$Z_s = 2.869 \text{ Log } \frac{V}{v} - .0788,$$ the values of $\frac{v}{V}$ being the reciprocals of the values in Column 8. The values, Z_g, are those given in Column 15.

The estimated revolutions are obtained from the equation

$$R_d = \frac{v \times 101.33}{\text{Pitch} \times (1-s)},$$ where v is taken from Column 4.

The estimated revolutions for these conditions are shown by Column 19, while in parallel, in Column 20, are shown the actual revolutions corresponding to the same speeds as taken from curve 20, Figure 1.

TO INSURE FREEDOM FROM CAVITATION IN THE DESIGN.

According to the theory of the method of design employed in the design of the propellers for the *Saratoga* and the *Lexington*, cavitation is divided into two types, cavitation due to excess tip speeds and cavitation due to excess thrusts. A third cause may be added, and that is cavitation due to bad blade sections. This latter is not however true cavitation, which is due to the limitations of the laws of nature but may be classed as purely mechanical, the bad sections producing excessive eddying in the water flowing around the sections and producing areas of excessive reduction of pressure in the wakes of the blades.

The theory employed fixes the limit at which cavitation due to excess tip speeds occurs at a speed of ship equal to the basic speed of the propeller. At all speeds in excess of this the propeller will be in cavitation.

The limit for cavitation due to excess thrusts for vessels of the *Saratoga* type is fixed by the following equation:

$$\frac{v}{V} = \frac{6\,(S^{1.4} + .236)}{\left(\frac{E.H.P.}{e.\,h.\,p._g}\right)^{.42} + 1.45},$$ where S is the basic slip and $\frac{e.h.p._g}{E.H.P.}$ the gross effective load fraction at the speed v.

In the cases of the vessels under discusison, as the power loss factor, K, equals unity, the gross and the net loads are equal at the speeds given in Column 4, Table 2.

This latter equation does not apply to all vessels however. In vessels of the tanker type, and vessels of finer types, where the buttock lines after swinging up from parallelism with the base line are practically straight instead of convex, the equation

$$\frac{\text{e. h. p.}_g}{\text{E.H.P.}} = \left(\frac{1 - S}{.79} \times \frac{v}{V}\right)^3 \text{ is used.}$$

Thus destroyers of the 1150 ton displacement type with straight buttocks built by the Bath Iron Works cavitate due to excess thrusts according to this last equation, while those of the same class with convex buttocks built by Fore River, cavitate according to the preceding equation.

Referring to Figure 3, the abscissae are values of the gross load fractions. With the scale of ordinates for $v \div V$, the Figure becomes the Cavitation Chart.

The original value of S used in the design was .177. On account of the expected reduced squatting, this value of S was not expected to be realized. However, by using it for design purposes the actual leeway before cavitation due to excess tip speeds would be encountered was increased over that shown by the design curves.

The estimated curve of $v \div V$ is shown dotted on the figure and is marked A, the point of design for full power, 180,000 S.H.P., is shown marked ⊙. The upper limit of safety is the horizontal line drawn through the ordinate value of $\frac{v}{V} = 1$. The limiting curve for cavitation due to excess thrusts for $S = .177$, is shown dotted, marked B, and cavitation due to this cause was not to be expected until the ship had been so loaded down as to reduce the speed fraction to .805, the corresponding speed, v, being then 27.15 knots.

Due to the lower and changing wake, however, the actual curve of $v \div V$ plots as shown by the full line marked C, while the curve of limitation for freedom from cavitation due to excess thrusts shifts to the full line curve marked D. The V (basic speed) has become 36.23 knots and that speed must be reached before cavita-

tion due to excess tip speeds can be expected, so long as the wake remains the same as at the actual design point, marked on C with ⊙. The speed at which cavitation due to excess thrusts may be expected has, however, risen to 27.53 knots.

If the limits for cavitation are correct, then the plotted curves of $v \div V$, both estimated and actual, give assurance that no cavitation will occur. The actual performances of the vessels as shown by their curves of shaft horsepowers and revolutions, are proofs that the expectations and the accomplished facts are in close agreement.

EFFECTIVE THRUSTS PER SQUARE INCH OF PROJECTED AREA. TIP SPEEDS.

In 1894, after the failure of the propellers of the British destroyer *Daring*, Mr. S. W. Barnaby, Chief Engineer of the Thornycroft Company, came to the conclusion that the trouble was due to what is now generally known as "cavitation," and fixed a value of 11¼ pounds per square inch of projected area as the maximum effective thrust which could be employed were cavitation to be avoided.

Rear Admiral D. W. Taylor, C. C., U. S. N., in his book on the "Speed and Power of Ships," comments on this limitation of thrusts as follows: "Greater thrusts have been successfully used and cavitation is liable to appear at much lower thrusts. . . . There is little doubt that the prime factors involved in cavitation are: (1) The speed of the blade through the water, which is evidently measured by the tip speed, and (2) the shape of the blade section."

By inspection of curve E, Figure 3, the curve of gross effective thrusts per square inch of projected area for the propellers of the *Saratoga* and *Lexington*, gross and net effectives being equal in this case, it will be seen how far from the truth was Barnaby, in fixing on 11¼ pounds as the limiting thrust. The maximum effective thrust was experienced with the *Saratoga*, which for the average speed of the five high runs, 34.64 knots, was 17.51 pounds.

Now with these vessels, propellers of the same diameters and projected area ratios can be designed which can be made to cavitate due to excess tip speeds or due to excess thrusts, at this same speed,

34.64 knots, and same effective thrust, 17.51 pounds, merely by lowering the pitch in the first case and raising it in the second. The plotted points for a series of such propellers, all driving the ship ahead at 34.64 knots, with constant engine power are shown by the line F, which is the line of constant effective thrust equal to 17.51 pounds. That propeller whose performance plots at F will be on the verge of cavitation due to excess tip speeds, while that which plots at H will be just entering cavitation due to excess thrusts. All propellers plotting between F and H will be in the safe range.

Taylor's statement as to the effect of tip speed is very general and offers no guide as to the limit.

According to the theory by which the propellers of these two vessels were designed, the actual tip speeds when just entering cavitation due to excess tip speeds are very closely approximated by the equation

$$\text{T. S.} = 38148\left(\frac{\text{P. A.}}{\text{D. A.}}\right)^{.8365 + 2\left(\frac{\text{P. A.}}{\text{D. A.}}\right)}$$

where $\frac{\text{P.A.}}{\text{D.A.}}$ = Projected area ratio of a three-bladed propeller, having projected area forms of the Standard Dyson Oval. This is basic, and equals ¾ total projected area ratio for a four-bladed propeller, the forms also being standard.

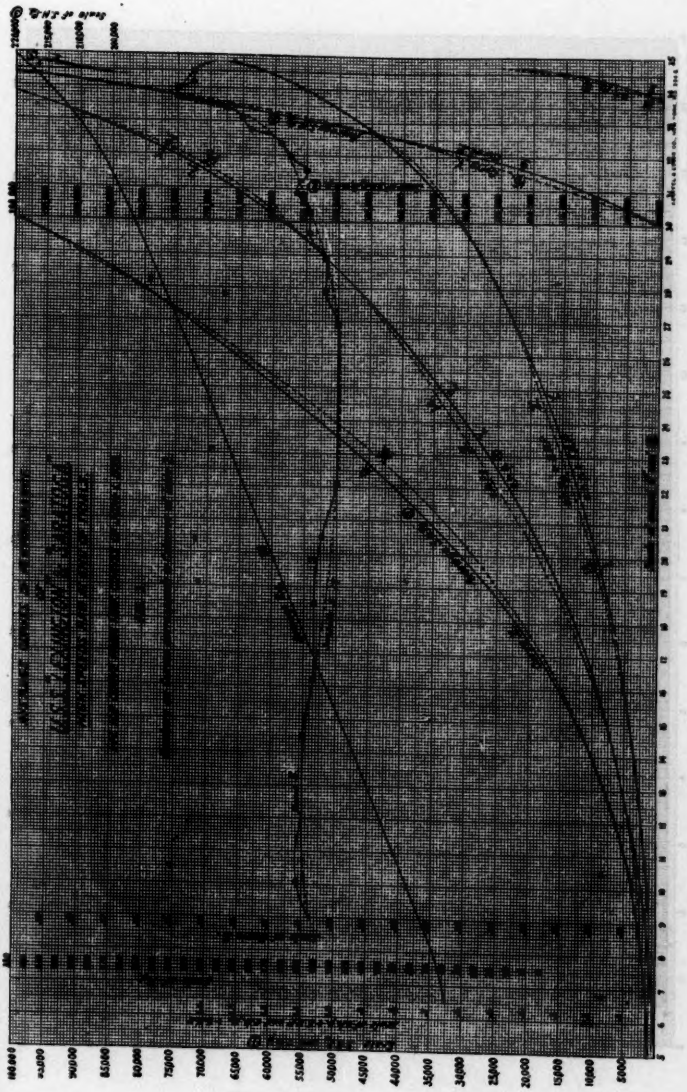
It will be noted that according to this equation, the tip speeds when the propeller is at the verge of cavitation due to excess tip speeds, depend solely upon the basic projected area ratio, $\frac{\text{P.A.}}{\text{D.A.}}$. As these basic projected area ratio forms are all standardized, Taylor's expression of "mean blade width" may be used and it can be stated, then, that the limiting cavitating speed depends upon mean blade width.

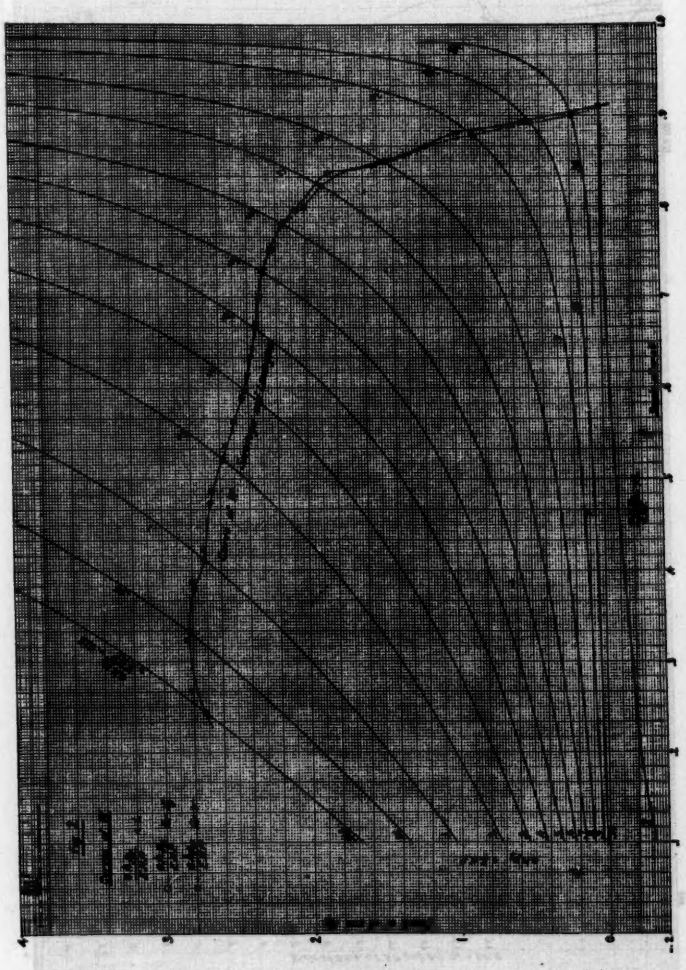
The curve of tip speeds, average for the two ships, is shown on Figure 3, Marked G. It will be seen that at the point of maximum performance the actual tip speed was 15,000 feet per minute.

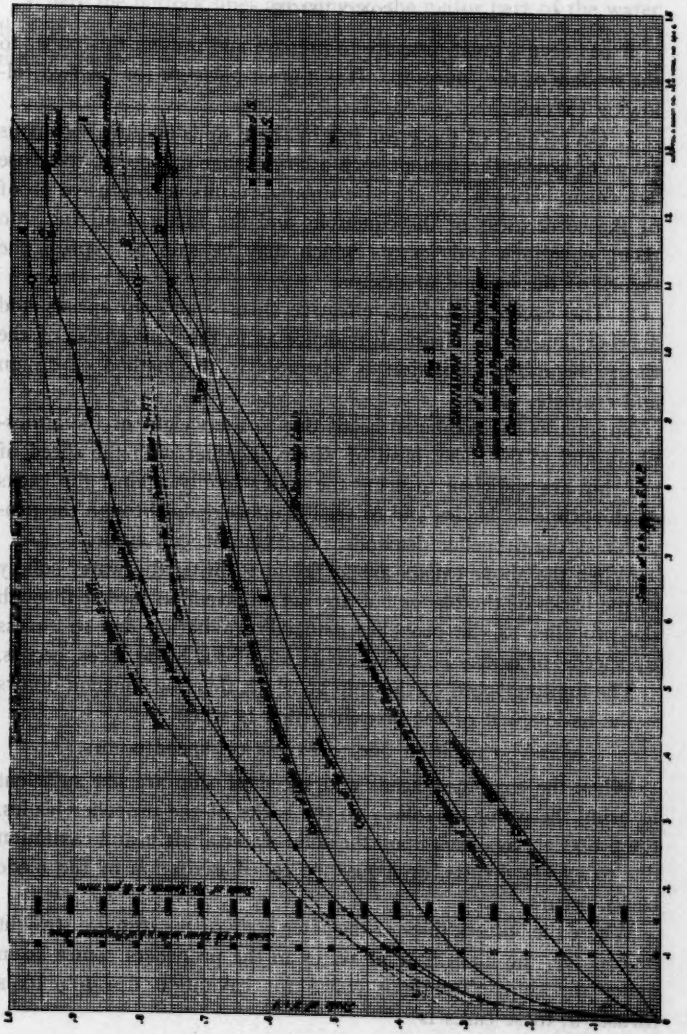
To furnish more information on the subject of gross effective thrusts and tip speeds and the relation to cavitation, Table 3 has been prepared and is here submitted.

TABLE 3. EFFECTIVE THRUSTS AND TIP SPEEDS

Vessel.	Slip B.C.	Proj. Area Ratio	$\frac{V}{V}$	Gross Wf. Thrust per □" P.A	Tip Speed	Cavitating and Type.
SARATOGA	.415	.65	.945	17.51	15000	No.
PENNSYLVANIA	.525	.359	.926	10.58	6866	No.
ARKANSAS	.52	.501	.8648	13.53	9980	No.
UTAH	.53	.556	.7021	13.06	9651	No.
PAUL JONES	.288	.43	.8917	9,863	7786	No.
PREBLE	.288	.358	1.011	11.42	7716	Slight. Tip Speed.
PERRY	.288	.275	1.055	11.34	6713	Yes. Tip Speed.
COLORADO (Old)	.485	.29	1.007	13.536	7046	Yes. Tip Speed.
WEST VIRGINIA (Old)	.485	.281	.9795	15.596	6926	No.
MONTANA	.49	.293	1.04	12.030	6954	Yes.
NORTH CAROLINA	.49	.29	1.018	13.368	7128	Yes.
BATTLE CRUISER	.450	.62	.609	14.45	9849	No.
COLUMBIA {Wing	.510	.257	1.046	10.402	6291	Yes.
COLUMBIA {Center	.563	.22	.9694	10.16	5617	No.
MINNEAPOLIS {Wing	.510	.237	1.034	10.086	6244	Yes.
MINNEAPOLIS {Cent.	.563	.22	.9809	12.31	5815	No.
IOWA	.628	.261	.9125	9.376	5879	No.
MAINE	.635	.267	1.05	9.908	5672	Yes. Tip Speed.
NEBRASKA	.65	.287	1.0075	10.828	6709	Yes. Tip Speed.
CHINESE G.B. {Wing	.498	.5	.7801	17.18	9675	No.
CHINESE G.B. {Cent.	.555	.5	.771	13.96	8905	No.
MONTGOMERY	.576	.243	1.036	8.472	6227	Yes. Tip Speed.
FULL BODY MODEL	.855	.267	.8076	7.096	4856	Yes. Ex. Thrust.
LAMBERTON	.32	.648	1.013	16.90	16050	Slight. Tip Speed.
NEW MEXICO	.55	.3567	.2795	1.848	3200	Yes. Ex. Thrust.
EAGLES	.465	.606	.4607	2.072	5645	Yes. Ex. Thrust.
DENVER	.499	.54	.5871	4.401	6587	Yes. Ex. Thrust.







CONCLUSION.

It has been a great pleasure to prepare and present this paper to the readers of the JOURNAL. It is a great pleasure for the two following reasons:

1. On account of the great size of these ships and the enormous range of power throughout such a large range of speeds, the close agreement between actual and estimated results is conclusive proof of the correctness of the equations and the factors entering into them, equations and factors which may be said to express the theory of the method of design.

2. The exceedingly high effective thrusts and tip speeds add another link to the chain of evidence as to the correctness of the theory regarding cavitation as advanced in the method of design and estimate, viz:

So long as the combined heads due to the weight of the atmosphere and to the mean head of water, together with the speed of the ship are sufficient to bring the water solidly against the backs of the propeller blades and to feed it unbrokenly through the propeller as the propeller demands, no cavitation can occur.

When the above causes are sufficient to bring the water solidly to the backs of the blades but are not sufficient to feed it through the propeller as the propeller demands, cavitation due to excess tip speeds, or it may be called cavitation of the blade faces, occurs.

The limiting speed of the ship at which this phenomenon occurs is the basic speed of the propeller, that is when

$v = V$, cavitation due to excess tip speed occurs.

When the combined heads and speed of ship are not sufficient to bring the water up solidly to the backs of the propeller blades, the propeller sags back upon the solid water abaft it and cavitation due to excess thrusts, or it may be called cavitation of the blade backs, occurs.

Where the buttock lines of the after body of the ship are straight or concave, the main flow of water to the propellers comes from under the ship. The limit curve for cavitation due to excess thrusts in these cases is

$$\frac{\text{e.h.p.}_g}{\text{E.H.P.}} = \left(\frac{1-S}{.79} \times \frac{v}{V}\right)^3.$$

Where the buttock lines are convex, the major part of the water flow to the propellers is from the sides. The limit curve equation then becomes

$$\frac{v}{V} = \frac{6\,(S^{1.4} + .236)}{\left(\frac{E.H.P.}{e.h.p._g}\right)^{.42} + 1.45}.$$

NOTES ON NAVY YARD MANAGEMENT.
FINANCIAL CONTROL.

By Captain H. G. Bowen, U. S. Navy, Member.*

1. The first and most important requisite of efficient Navy Yard Management is adequate financial control. This is true, irrespective of the type of management involved and the system described herein can be employed in any Navy Yard.

2. The expression, "financial control," is used in this discussion to designate the methods employed:

(*a*) to determine the funds required to carry out the overhaul schedule according to estimates;

(*b*) to allocate or budget the funds by ships and other activities after such funds have been allotted by the Bureaus;

(*c*) to determine the number of men required to meet the Yard's commitments in regard to Fleet work and New Construction;

(*d*) to determine correct applied shop rates if and when current applied shop rates are observed to be inappropriate;

(*e*) to budget shop expense.

3. Since the official designation of officers performing Industrial duties at Navy Yards differ, it is desired to digress briefly in order to define the duties of the officials of the Puget Sound Navy Yard, who are responsible under the Manager and the Commandant for the control of finances in the Industrial Department.

4. The Chief Production Officer is in charge of the Production Division, which includes all shops, inside and out, except X70, the building trades shop, which is under the Public Works Officer. The Chief Production Officer performs the duties which, prior to General Order No. 53, were performed by the Outside Superin-

* Chief Production Officer, U. S. Navy Yard, Puget Sound.

tendents and Shop Superintendents of both the old Hull and Machinery Divisions. In addition, he is in charge of all expenditures, direct or indirect, of the following appropriations:

501 Construction and Repair,
601–1 Engineering (less radio),
601–2 Engineering Radio (except for shore stations),
401 Ordnance and Ordnance Stores,
750–Z Naval Supply Account, Manufacturing,
309 Instruments and Supplies,
851–3 Aviation, Navy, Maintenance and Repairs, Yard,
161 Increase of the Navy,
777–18 Naval Working Fund, Other Government Departments,
322–4 Organizing Naval Reserve Force, Engineering,
322–5 Organizing Naval Reserve Force, Construction and Repair.

5. The Planning Division prepares the requests to the Bureaus for funds after having satisfied the Chief Production Officer that the amounts to be requested are adequate to meet the requirements of the Production Division. The Chief Production Officer is responsible for the allocation and expenditure of funds after they are authorized. He is charged with not exceeding the Bureau allotments and with meeting scheduled dates of completion of vessels under overhaul and under construction. He is charged with the administration of the funds expended for direct shop expense.

6. Thus it will be seen that the Chief Production Officer is in charge of the Financial Control of the Industrial Department. What this Financial Control involves, we will now proceed to examine:

DIVISION OF BUREAU ALLOTMENTS, INTO LABOR, INDIRECT, AND MATERIAL.

7. After the Bureau allotments are received, they are first divided among labor, indirect, and material in accordance with the following example:

Total approved C&R (501 Less K),	
Allotment authorized for December	$120,000
Material requirements for December,	
Est. based on survey of outstanding work load	$14,230
Classified service, fixed charge	13,660
Indeterminate docking, based on prospective docking programme	3,600
Total	$31,490

Note: The material requirements for this appropriation usually approximate 21 per cent of the direct labor. This amount was compared with the material requirements indicated on the form, "Financial Analysis," under the heading, "December Obligations." See paragraph 25.

Total approved allotment	$120,000
Material, Classified Service, etc.	31,490
Labor and Indeterminate	$88,510

8. Since indeterminate has been running 38 per cent for this

Appropriation $\frac{\$88,510}{1.38}$ =	$64,140	Direct Labor
	13,660	Classified
	$77,800	Total Labor
	27,970	Indeterminate (Includes Dry Dock Indeterminate)
	14,230	Material
	$120,000	

9. The indeterminate, $27,970, is then split for accounting purposes according to the ratio determined by the Accounting Office for each month. For December 82 per cent was added to labor and 18 per cent to material.

Total Labor	$77,800
82 per cent Indet.	22,935
Labor and 82 per cent Indet.	$100,735

Material	$14,230
18 per cent Indet	5,035
Material and 18 per cent Indet	19,265
Total Allotment	$120,000

10. The Accounting Office is informed of how the allotments have been divided and they are taken up on the Accounting Office Forms accordingly:

PREPARATION OF MONTHLY BUDGET, OF FUNDS BY SHIPS AND OTHER ACTIVITIES.

11. The first problem which confronts the Industrial Department at the beginning of a month is the allocating or budgeting of funds and the determination of the number of men required to make scheduled progress on vessels of the Fleet under overhaul and vessels under construction.

12. A typical solution of this problem is submitted for the month of December 1928.

13. "From: Chief Production Officer.

To: Production Division.

Subject: Budget of funds by Ships and Other Activities — December (Direct Labor only).

"1. The distribution and expenditure of funds and distribution of men during the month of December 1928 will be as follows, based on available funds and prospective work load:

Shop Production Officer.

Supervision	70 men per day
Apprentices	10
Other Shop Expense	100
Clerks	23
Title Z Shops	76
Leave with pay	153
Leave without pay	30
	462

Productive Labor	1592
Total	2054
On rolls 11-26-28	2238
Apparent labor surplus	184

750–Z Manufacturing.

$10,500 or $420 per day or 64 men. (Unit Wage Factor 6.58.)

"2. *Hull Production Officer.*

501 Except K.		*Total Direct Labor*		
Decatur	7 days	$1917	$274 per day or	43 men
Cuyama	14	7000	500	78
Moody	15	4648	303	47
Zeilin	20	5929	297	46
Colorado	25	31452	1257	197
J. F. Burnes	8	1500	188	29
New York	1	200	200	31
Somers	1	100	100	16
Misc. Ships	25	3000	120	19
Yd. Craft & Voy. R.		1600	64	10
Ships Out of Com.		3700	148	23
Other G		1600	64	10
Title V		1000	40	6
Title E		824	33	5
Title C		230	9	1
		$64700	Ave. $2590 per day or	406 men
Classified		13660		
Total		$78360		
Allotment		78360		

(Unit Wage Factor 6.39.)

501–K.

Decatur	7 days	$30	$4 per day or	1 man
Cuyama	14	1095	78	12
Moody	15	3144	210	33
Zeilin	20	3858	193	30
Colorado	25	2300	92	14
J. F. Burnes	8	300	38	6
New York	1	100	100	16
Somers	1	0	.....	....
		$10827	Ave. $433 per day or	68 men

(Unit Wage Factor 6.37.)

Cruiser No. 28, C&R.

25 days. $56750 or $2270 per day or 355 men.

(Unit Wage Factor 6.39.)

"3. *601–1. Engineering and Ordnance Production Officer.*

Decatur	7 days	$2895	$413 per day or	65 men
Cuyama	14	9408	672	106
Moody	15	10995	733	116
Zeilin	20	12841	643	102
Colorado	25	17657	707	112
J. F. Burnes	8	3500	438	69
New York	1	200	200	32
Somers	1	100	100	16
Misc. Ships	25	425	17	3
Yd. Craft & V.R.		1890	76	12
Ships Out of C.		1890	76	12
Other G		1420	57	9
Title V		650	26	4
Title E		1600	64	10
Title C		330	13	2
Title Z		500	20	3
		$66301	Ave. $2650 per day or	420 men
Classified		9420		
Total		$75721		

(Unit Wage Factor 6.32.)

Cruiser No. 28, Engineering.

25 days. $15620 or $625 per day or 99 men.

(Unit Wage Factor 6.32.)

401.

Decatur	7 days	$710	$101 per day or	16 men
Cuyama	14	387	28	4
Moody	15	2244	150	23
Zeilin	20	3038	152	24
Colorado	25	11256	450	70
J. F. Burnes	8	900	113	17
New York	1	100	100	16
Somers	1	100	100	16
Ships Out of Com.	25	700	28	4
Other G		1000	40	6
Title V		290	12	2
Title Z		290	12	2
		$21015	Ave. $842 per day or	131 men
Classified		815		
Title G. for Supply Dept.		1000		
Total		$22830		
Unobligated (Title P)		4585		
Allotment		$27415		

(Unit Wage Factor 6.44.)

"4. *Miscellaneous Appropriations.*

309, 601–2, 851–3, 201, Misc. — 25 days. $7800 or $312 per day or 49 men.

(Unit Wage Factor 6.37.)

"5. *Force Computation for December, 1928.*

(25 Working Days.)

Appropriation	*Productive Labor*	*Wage Factor*	*Average Funds per Day*	*Average Men per Day*
501 except K	64700	6.39	2590	406
501 K	10827	6.37	433	68
601–1	66301	6.32	2650	420
401	21015	6.44	842	131
750–Z	10500	6.58	420	64
Misc. Appro.	7800	6.37	312	49
Cruiser No. 28, C&R	56750	6.39	2270	355
Cruiser No. 28, Eng	15620	6.32	625	99
Total	253513	6.37	10142	1592

Supervision	70
Apprentices	10
Other Shop Expense	100
Clerks	23
Title Z Shops	76
Leave with pay	153
Leave without pay	30
	462
Productive Labor	1592
Total	2054
On rolls 11-26-28	2238
Apparent labor surplus	184

H. G. BOWEN."

14. A discussion of certain features of the monthly budget letter is now in order.

SHOP PRODUCTION OFFICER.

15. The number of men assigned the Shop Production Officer for supervision, instruction of apprentices, "Other" shop expense, etc., is estimated from past performance, with an eye toward any marked change in the amount of the work load for the ensuing month. "Other" shop expense refers to all men charged to Shop expense and not otherwise accounted for in this enumeration.

750–Z MANUFACTURING.

16. It has been estimated from the work on hand and from previous experience that $10,500 is a sufficient allowance for direct labor for 750–Z Manufacturing and based on an average wage (unit wage factor) for this class of work of $6.58 per day, 64 men can be employed under this heading.

HULL PRODUCTION OFFICER.

17. The table under this heading shows amounts allotted by the Chief Production Officer under direct labor, Appropriation 501 Construction and Repair, for various ships and activities. These amounts are determined by the necessity of meeting the dates of completion of vessels that complete within the month and by the necessity of making a pro-rated amount of progress on ships which complete in the next or following month, as well as keeping up with the work required by other activities. It has been determined that the average wage (unit wage factor) per day for work under 501 titles except K averages $6.39 per day. Hence the number of men required under this heading is 406.

18. Similar remarks apply to the table under "501–K," "601–1," "401," and "Miscellaneous Appropriations."

19. It will be noted that adding together all the numbers of men who can be employed under specific appropriations gives the predicted productive labor in terms of the number of men, which is 1592 in the example of paragraph 13. This total added to the number of men assigned to overhead and to Title Z Shops, and including clerks (462 men), gives the number of men that can be employed, i. e., paid, and the differences between that sum, 2054, and the total number of men on the rolls of the Production Divi-

sion, 2238, gives the number of men to be laid off, 184. The lay-off was apportioned among the various shops in detail by the following method.

20. The record of five months showed that work under each appropriation was split between Hull and Engineering shops according to the following percentages:

	Hull Shops per cent	*Engineering Shops per cent*
501 except K	71.0	29.0
501–K	66.0	34.0
601–1	16.5	83.5
401	9.0	91.0
750–Z	76.0	24.0
Miscellaneous Appropriations	41.0	59.0
Cruiser No. 28 (C&R Eng.)	75.0	25.0

21. It is evident that the number of men enumerated in the table under the heading "Force Computation," 1592, must be split into Hull and Engineering groups in accordance with the above percentages which give us the following distribution:

	Average for December		
	Total Hull & Eng. Shops	*Hull Shops*	*Engineering Shops*
501 except K	406	288	118
501–K	68	45	23
601–1	420	69	351
401	131	12	119
750–Z	64	49	15
Miscellaneous Appropriations	49	20	29
Cruiser No. 28, C&R	355)		
Do Eng.	99)		
	1692	824	768
	Hull	*Eng.*	*Total*
Productive Labor 11-26-28	884	892	1776
Estimated Do for December	824	768	1592
Necessary lay-off	60	124	184

22. The Engineering Production Officer and the Hull Production Officer were then instructed to lay off 124 and 60 men, respectively. The Shop Production Officer was directed to make a corresponding cut in supervisory force and in "Other" shop expense. No difficulty is experienced in pro-rating the lay-off among the individual shops on the basis of shop strength and taking into consideration any marked change in the character of the work load affecting trades.

23. The two fundamental forms or reports most essential for financial control are the "Daily Distribution of Personnel by Appropriations" Sheet and the "Daily Financial Analysis" Sheet.

DAILY DISTRIBUTION OF PERSONNEL REPORT.

24. The "Daily Distribution of Personnel" Sheet, shown as Figure 1, needs no extended explanation or comment. It enables all hands to know how labor is distributed, by appropriations, on ships, on shop account numbers, and on other activities. It permits the calculation of the average daily wage by appropriations, approximate daily expenditures for labor in advance of returns from the Accounting Office, together with predictions for productive labor, and the over and under absorbed, Indeterminate Expense.

FINANCIAL ANALYSIS REPORT.

25. The "Daily Distribution of Personnel" form also acts as a feeder form for the "Financial Analysis" report, shown in the case of appropriation 601–1 Engineering. This form is shown as Figures 2 and 3, Figure 2 for direct labor and Figure 3 for material. Referring to Figure 2, it will be noted that in the columns headed "Predicted Expenditures to End of Month" are recorded in the case of each ship, the number of men working on the ship on that date (from Figure 1), the number of working days remaining in the month or in the overhaul period, for vessels which complete during the current month, and the average wage, the product of which gives the predicted expenditures for the remainder of the month. The column headed "Allotment" is obtained after dividing the Bureau's allotments into Labor, Indirect, and Material, and further sub-dividing into allotments by ships and by projects as

shown in the Monthly Budget letter, paragraph 13. The entries in the column headed "Released" indicate the total amount of the direct labor of the job orders released by the Chief Production Officer. The column, "Estimated Total," indicates the plans of the Chief Planning Officer in regard to estimates. The information for the columns, "Total Expended to Date," and "Expended this Month," are of course obtained from the Accounting Office. The entries in column, "Total Predicted Expenditures for this Month" are the sum of entries in columns "Expended this Month" and similar entries in column "Predicted Expenditures to End of Month." In the case of activities other than ships, i. e., work on standing job orders, as Yard Craft, Decommissioned Ships, Titles N, V, C, etc., they represent the allotments of the third column remaining unexpended. The entries in columns under "December Obligations" and "January Obligations," are obtained by subtracting the progress indicated for the current month from the total estimate. For ships which have not yet arrived, the entries represent an average daily rate of progress, on work available in advance of arrival.

26. Attention is invited to the index numbers which indicate the progress made on the work program. Over 100 indicates that the work performed is ahead of schedule and under 100, that the work is behind schedule.

27. The Financial Analysis form is really the key report of the Production Division. It indicates to the Ship Superintendent as well as to the Chief Production Officer and to the Manager the progress of work. It puts a brake on the issuing of job orders by Planning Division since the Chief Production Officer is empowered to hold up the release of job orders until a satisfactory balance is obtained. It maintains a running total of the "obligations" for the two ensuing months and this information enables the Chief Production Officer to keep a check on the amount of funds or monthly allotments to be requested from the Bureaus.

FORCE DISTRIBUTION SHEET.

28. Figure (4), the "Force Distribution Sheet" (for November) is a handy reference of certain vital information abstracted from Figure 1. The "standards" are obtained from the "Month-

ly Budget Letter" (see par. 13 of this article). It should be noted, however, that the Budget Letter is for December and not November.

SHOP BUDGETS.

29. Another important monthly computation is the determination of the monthly shop budget. As indicated under force computation for December 1928, the Productive Labor was estimated as $253,513 (see par. 13). The computation follows:

Productive Labor less Shop X70 (see par. 13, sub-par. 5)	$253,513
Productive Labor X70 (same as November)	19,000
Total Productive Labor	$272,513
Overhead absorbed at estimated rate of 57.5 per cent	$102,000

ESTIMATED OVERHEAD FOR DECEMBER.

"Other Shop Expense"	$38,700
Power Transfer	12,800
Leave, less X70 (1562 men $\times$ 1/10 $\times$ $7.32)	11,380
Holiday, less X70, December 25, 1928	11,600
Shop X70 Leave and Holiday	2,500
General Expense Transfer	20,000
Total estimated overhead	$96,980
Estimated over absorbed for December (102,000 — 96,980)	5,020

30. Hence the estimated actual overhead rate is

$$\frac{\$96,980}{272,513} = 35.6 \text{ per cent}$$

The following questions naturally arise: Why was "Other" Shop Expense set at $38,700? The value of this item from its nature varies with the number of working days in the month. When under close scrutiny and after shop extravagance has been eliminated and shop equipment overhaul has been adequately organized,

there will be little variation from any other cause. Power Transfer and General Expense Transfer do not vary through large limits in an economical administration, unless in the case of Power Transfer, there should be an abnormal increase or decrease in power output, with no change of rates. Leave is estimated at the normal or average amount of leave for those who have been in service for a year or more. X70 is not attached to the Production Division and its values of productive labor, leave and holidays are based on past averages and present strength.

31. The Budget of $38,700 is distributed to the Shops in the following manner:

DISTRIBUTION OF OTHER SHOP EXPENSE.

32. There are 174.5 working days in the 7-month period under consideration, December to June, inclusive. The daily estimated expenditures for "Other Shop Expense" are $\frac{270,000}{174.5} = \$1548$ per day. The sum of $270,000 represents the average value of "Other Shop Expense" for a 7-month period. The estimated "Other Shop Expense" for the 25 working days of December is therefore $\$1548 \times 25 = \$38,700$. The budget of $38,700 is distributed to the various shops as follows:

X11	$5800
X17	2100
X23	1150
X26	2250
X28	310
X31	6700
X38	4470
X41	1260
X51	2610
X53	980
X56	2400
X61	750
X63	570
X68	1000

X70	2050
X71	990
X72	1912
X73	214
X74	409
X93	208
X94	567
Total	$38700

The above distribution to the various shops is based on the percentage of the total Other Shop Expense incurred by each shop during the same period (December to June, Inc.), of the last fiscal year.

"OTHER SHOP EXPENSE," ESTIMATED BY SHOP MASTERS.

33. On or about the 15th day of each month each Master or Supervisor in charge of a shop is required to submit on a form prepared for that purpose, Figure (5), an estimate of the minimum expenditures required under each of the following shop expense accounts for the efficient conduct of the work in prospect for the month: 01, 02, 03, 11, 12, 14, 15, 3, 40, 41, 5, 6, 7, 8 and 9. In preparing these estimates, Supervisors in Charge of Shops are required to anticipate expenditures for Consumable Supplies, Fuel, Miscellaneous Labor, Supervision, also for repairs and replacements incident to normal wear and tear on plant facilities, tools and equipment. The estimates when submitted are carefully checked against the budget prepared by the Chief Production Officer. In case the estimates exceed the budget, the Shop Supervisor concerned is conferred with and means are devised to provide the additional funds when in the interests of economy such a course is considered necessary.

DAILY CHECK ON EXPENDITURES.

34. A daily statement of expenditures for "Other Shop Expense" accounts is prepared on the Hollerith Machines by the Accounting Office for use in checking expenditures against the budget. This statement is, as a rule, available by noon the day

following that upon which the charges were made. This statement, in conjunction with the monthly budget, is used as a guide by the Shop Production Officer in considering all work orders and stub requisitions covering Labor and Materials chargeable to Shop Expense Accounts which are submitted by the Master of the Shops for approval by the Shop Production Officer.

35. The method of establishing overhead rates for as long a period of time as possible will always engage the interest of Navy Yard administrators. The following example is an illustration of how the present rates of the Puget Sound Navy Yard were determined. It was desired to have the new rates effective from 1 December 1928, and as it was then the middle of November, it became necessary to estimate the over or under absorbed, for November. This was estimated at $40,000, over absorbed and added to over absorbed at the end of October, $136,426, gave a predicted total over absorbed for the beginning of the new rate period of $176,426.

36. An examination of the Leave situation is given below:

LEAVE, PRODUCTION DIVISION, FISCAL YEAR 1929.

Month	*No. Working Days*	*Man Days Leave Taken*	*No. of Men on Roll*	*No. of Men Entitled to Leave*	*No. of Men Days Leave Accruing Each Month*
July	23	2930	1799	1351	3110
August	25	2929	2085	1452	3630
September	21½	2198	2271	1543	3320
October	27	2852	2413	1562	4220
November	25	2310 Est.	2374 Est.	1562	3910
		13219			18190
Total Man Days leave taken					13219
Total Man Days leave not taken					4971
December	25		1984 Est.	1562	3910 Est.
January	26		1984 "	1610	4190 "
February	23		2038 "	1610	3700 "

March	26	2077 Est.	1610	4190 Est.
April	26	2112 "	1610	4190 "
May	26	2144 "	1610	4190 "
June	22½	2188 "	1610	3620 "
Total leave accruing 1 Dec. 1928 to 30 June 29....				27990 "
Total leave remaining unliquidated 1 Dec. 28.....				4971
Total leave				32961

37. Holidays (Unit Wage Factor $6.66).

	Roll	*Deduction*	*No. of Men on Holiday*	*Cost*
Christmas	1940	200	1740	$11,600
New Year	1940	200	1740	11,600
Washington's Birthday	2058	200	1858	12,350
Memorial Day	2168	200	1968	13,100
5 Half Holidays	2214	200	2014	33,400
				$82,050

Note: The deduction of 200 men includes the number of men usually on leave without pay, clerks, and employees of Title Z Shops.

	Prod. Labor No. of Men	*Non-Prod. Labor No. of Men*	*Roll*	*X70 Prod. Labor*	*Total Prod. Labor*
November	1952	430	2382	$19,000	$329,000
December	1522	418	1940	19,000	261,000
January	1589	418	2007	19,000	283,000
February	1638	420	2058	19,000	259,000
March	1677	420	2097	19,000	298,000
April	1712	422	2134	19,000	303,000

May	1744	424	2168	19,000	309,000
June	1788	426	2214	19,000	276,000
					$1,989,000

Note: Example, 1952 × 25 × $6.37 plus $19,000 = $329,000.

Note: The average number of men employed on productive labor other than new construction is 1130. To this has been added the number of men required for new construction according to the Cruiser man power curve. The total being the Productive Labor indicated.

38. Budget of estimated expenditures for 7 months ending June 30, 1929.

	1 Dec. to 30 June
General Expense Transfer	$139,500
Power Transfer	90,000
Other Shop Expense	270,000
Leave Production Division	241,000
Holiday and Half-Holiday Prod. Div.	82,050
Holiday and Half-Holiday and Leave X70	22,287
Total estimated Shop Expense	$844,837
Total estimated over absorbed 1 Dec. 1928	176,426
	$668,411
Desired over absorbed 30 June 1929	80,000
To be absorbed 1 Dec. 1928 to 30 June 1929	$748,411

New Shop rate $\frac{\$748,411}{1,989,000} = 37.6$ per cent.

39. After conference with the Accounting Division and having in view work load prospects for each shop as well as the over and under absorbed of each shop, the total productive labor was divided among the various overhead shops and each shop rate was assigned in accordance with the following tables:

ESTIMATED PRODUCTIVE LABOR.

Shop	*One Month*	*Seven Months*	*Rate Per Cent*
X11	51,050	357,350	30
X17	12,000	84,000	40
X23	3,500	24,500	80
X26	10,000	70,000	40
X28	1,300	9,100	50
X31	40,000	280,000	45
X38	40,000	280,000	30
X41	10,000	70,000	50
X51	20,000	140,000	40
X53	6,000	42,000	50
X56	18,500	129,500	35
X61	5,000	35,000	40
X63	4,000	28,000	60
X68	8,500	59,500	35
X70	20,000	140,000	33.3
X71	8,000	56,000	40
X72	17,000	119,000	40
X73	1,800	12,600	35
X74	2,000	14,000	40
X93	600	4,200	50
X94	4,750	33,250	40
	284,000	1,988,000	

40. It must be evident to any one who has had duty in the Industrial Department of a Navy Yard that some such system of financial control is absolutely necessary if ships are to be completed on time and the Bureau allotments not only not exceeded but intelligently requested.

41. This system herein described does not supplant the Accounting System, it supplements it. The Accounting System accurately depicts the past. This financial control ties up the past with the present and the future. Instead of competing with the Accounting System, it provides a means for making the fullest use of the Accounting System.

CONCLUSION.

42. It is now pertinent to consider what has been accomplished to justify the additional time and effort involved in putting into effect and maintaining this financial control system:

(*a*) Funds from the Bureaus have been more accurately requested than heretofore.

(*b*) The budgeting of funds by ships and other activities has permitted a close supervision of expenditures for these purposes and therefore has undoubtedly effected a reduction in cost.

(*c*) The laying off or taking on of men has been expeditiously handled according to plan. By keeping the number of men on the roll at a minimum, there is always on hand more money than required for the pay roll. Thus hurried lay offs and calls and the sudden forcing of a large number of men on leave with or without pay have been avoided. All of which has reduced the labor turnover.

(*d*) In determining correct applied shop rates, it has been necessary to map out future costs and expenditures. Actual deviations from the plan are immediately noted and the necessary corrective action immediately taken.

(*e*) The budgeting of shop expense has furnished an effective instrument for the administration of shop finances. It has mainly contributed to the reduction of shop overhead from an average of 55 per cent in the Spring of 1927 to an average of 37.5 per cent for the period beginning 1 December, 1928, without any sacrifice in upkeep and maintenance of equipment.

Leave and holiday, power and general expense transfers are not entirely under the cognizance of the Chief Production Officer. Eliminating these charges for purposes of comparison, the average overhead for a period of ten (10) months ending April 30, 1927, was 21.47 per cent, and for a period of four (4) months ending October 31, 1928, was 13.17 per cent, a difference of 8.3 per cent. The total productive labor for the period May 1, 1927, to October 31, 1928, was $3,842,463.22. The actual shop expense for this period, leaving out the same charges referred to above, was $607,921.55. If the same Shop Expense rate, namely = 21.47 per cent, had obtained during this period, the actual Shop Expense

would have been $824,976.86. This indicates for the same period, a saving in Shop Expense of $217,055.30.

If the actual rate of saving 8.3 per cent be applied to the estimated monthly productive labor for the seven months period beginning 1 December 1928, an apparent saving of about $24,500 per month is indicated and probably will be realized.

Shop expense cannot be expected to vary directly with productive labor, even under ideal circumstances, so that any Shop Expense computations based on a period where productive labor fluctuates, are open to criticism as to their exactness, but the quantities involved in these computations are so large that none can deny that a large amount of money has been saved by practicing rigid shop economy through the agency of a shop budget.

43. The author of this paper wishes to acknowledge with thanks the assistance rendered by the following in the development of the financial control system: Lt. Comdr. C. F. Osborn (C. C.), U. S. N., Mr. G. A. Hastings, Mr. H. A. Babcock, Mr. R. W. Clark, Mr. E. L. Comings, all of the Puget Sound Navy Yard.

SHIPS UNDER

Date Nov. 26 No. 6

		Hull Production Hours										
		11	17									
(1) DECATUR	401						6					
	401	4	2		2		2		3	5	4	
	401K											
	401	4			2		6					
	Mixed											
(2) MOODY	401											
	401	5	13		8							
	401K	16							1			
	401	8		4			7					
	Mixed											
(3) SELLIN	401											
	401	6	3							8	8	
	401K	15	3						2			
	401	5					7					
	Mixed											
(4) ONTARIO	401											
	401	26	2			1	10	4		3	16	
	401K	6	7									
	401	3					6		1			
	Mixed				1							
(5) COLORADO	401						14					
	401	19	6				21	2		12		
	401K											
	401		2									
	Mixed											
(6) TATNUCK SWALLOW PANTUCKET MANOPAC	401											
	401	6	5			7		2	2			
	401K											
	401		2									
	Mixed											
(7) NEW MEXICO IDAHO SARATOGA TENNESSEE	423								2			
	601											
	601	161-A					5					
	601											
	Mixed											
	401											
	401											
NEW CONST'N (10) CRUISER No. 11	May-3											
	Oct-5	1943 107										
	Dec-5	1944 181		2		5	7		4		3	
	Aug-4			12					8			
TOTALS		413	45	18	7	13	91	8	17	28	85	

PS UNDER OVERHAUL

					Machinery Protection Ships										
		2		6	10	6		9				1		26	36
5	4			22	3	5		17	2	12		1		39	61
				12	41	20	16	9	2	6			1	105	117
								3						3	3
					11	10		5						26	26
				20	3	4		1						8	28
				19						6				6	25
				19	52	50	2	7		4		2	3	120	139
								7						7	7
					7	6						1		14	14
8	2			19		4		1		2				7	26
				20											20
				12	20	14		3	1					38	50
								3						3	3
						4								4	4
3	16			68	9	4		1						14	76
				13					1	2				3	16
				10	77	44	10	7	6	9		3	1	157	167
				1				1						1	2
				14	2	18								20	34
12		2		62		12			2	34				48	110
				2	2	4	2	11		2				21	23
				22	2					5				7	29
				2	10	16	3	16		2				49	51
								1						1	1
				2											2
		2		7				6	2					8	15
					2									2	2
		3		309	2		2	1				9		15	324
				14	7				16	1		6	2	32	46
28	25	6		671	261	234	35	109	32	85		23	7	786	1457

(Fig. 1(a))

Form Guild Pupil Board Shop Work Admin (N)

PRODUCTION

Daily Distribution of P

Date Nov. 26 1918 No. 8

	Hull Production Shops									
	11	17	23,27	29	61	73,85	75	89	95	71,99
1. Increase of Navy (101)	288		14		5	12		6		3
2. Yards and Docks (201)	2	2							1	
3. Improvements and Repairs (300) — Regular				1						
3. Improvements and Repairs (300) — Projects										
4. Naval Militia, C & R (300-A)										
5. Naval Militia Exp. (300-B)										
6. Ordnance (301)						20		8		
7. C & R Projects (301 K)	39	10						3		
8. C & R—All Titles except (301 K)	82	34	5/7	4	24	111	6	79	36	47
9. Engineering (311 Subhead 1)	22	8	4	2		26		1		1
10. Radio, Naval, (311 Subhead 2)		3								
11. Supplies & Accts. (312, 314, 315)	3	14	16				1	5	15	8
12. Other Gen. Depts. (317-18)										
13. Special Depots (319-20)										
14. Aviation (321)										
15. Miscellaneous Appropriations	6					1				
16. Subordinate Shop Expense	36	29	13	2	8	19	2	7	9	4
17. Leave Without Pay	9	3				2			1	1
18. Non-Productive Labor	52	34	19 2/7	2	2	23	2	8 5/7	11	7
19. Productive Labor	435	69	28	7	29	168	9	23	53	51
20. Clerks and Checkers	4	2	5/7			2		2/7	1	1
21. Title 3 Shops	3		6					1		1
22.										
23.										
24. Shop Mimeo Roll	487	103	47 5/7	9	31	191	11	31 5/7	64	58

(Continued from

Non-Productive Labor (Classified Force + Men on Leave Without Pay + Subordinate Shop Expense (including Men on Leave W

Remarks:

CTION DEPARTMENT

ion of Personnel by Appropriations

					Machinery Production Shops											
68	71&W	74		Total	21	38	41	51,52	53	59	61	64	66	Total	Grand Total
	3	2		330	10		2	7	16	1		15	2	55	385
1				5			7					1		8	13
				1				6						6	7
		2		89	30	52	1/2	14			1/2	2		99	123
				52					1	6				9	61
38	47	2		358	18	35		23	4	57		1	2	130	488
	1			64	205	178	37	61	11	28		8	8	536	600
				3				10						10	13
15	8	6		62	20	2		1	3	2	65			93	155
					3									3	3
				7	2		5	3						10	17
9	4	2		185	48	24	5	33	6	46	5	3	41	211	336
1	1	1		17		6	2			1	1			10	27
11	7	3		164	51	31	7½	37	6	48	7½	3	43	236	462
53	51	12		884	285	266	51	121	37	95		27	10	892	1776
1	1			11	3	1	1/2	4		1	1/2		2	12	23
	1			11							65			65	76
64	56	15		1046	336	297	58½	358	43	143	71½	30	53	1180	2226

(Continued Summary on Sheets [illegible])

Men on Leave With Pay(+)Employees Engaged on Foreign Work, Outbuilding, or Manufacture of Poles and Cars

(Fig. 1 (b))

SHIPS UNDER OVERHAUL AND C

Date Nov. 26 No. 8

GENERAL AND SHOP EXPENSE

	Hull Production Steps										
	11 35,36	17	35,37	36	61	70,61	71	69	66	71,69	71
Totals from Page 1	413	45	16	7	13	91	8	17	28	25	6
(16) G-60 Groups, Gangways, etc. 601											
(16) G-60 Groups, Gangways, etc. Misc.											
(17) G-61 Groups, Ramps, etc. 611					11	14					
(17) G-61 Groups, Ramps, etc. Misc.											
(18) G-62 Groups, Transportation, etc. 621									10		
(18) G-62 Groups, Transportation, etc. 625											
(18) G-62 Groups, Transportation, etc. Misc.											
(19) G-64 Groups, Cranes, etc.	4	2	5/7			2		2/7	1	1	
(20) Cruiser Ship No. 1 [redacted]	9				2	46		2		14	
(20) Cruiser Ship No. 1 [redacted]											
(21) Ships Gun Cranes. [redacted]											
(21) Ships Gun Cranes. [redacted]	3				3	11				11	
(21) Ships Gun Cranes. [redacted]											
(22) All Other G Groups. [redacted]											
(22) All Other G Groups. [redacted]											
(22) All Other G Groups. Misc.	2										
(23) Title X. [redacted]	2	1									
(23) Title X. [redacted]		4									
(23) Title X. Misc.	6	2									
(24) Title V. [redacted]											
(24) Title V. [redacted]											
(24) Title V. [redacted]						3					
(24) Title V. [redacted]											
(25) Title S. [redacted]											
(25) Title S. [redacted]											
(25) Title S. [redacted]											
(25) Title S. 715S	3	14	16				1	5	15	8	6
(26) Other Shop Expense. Own Shop	4	4	2			1		1	1	1	1
(26) Other Shop Expense. Other Shop		4	8			6		3	3		
(27) Miscellaneous Expense		3				1					
(28) Z-0 Groups, Supervision, etc.	16	4	2	1	1	4	1	1	3	3	1
(29) Z-10 Groups, Apprentices, etc.											
(30) Z17, Z-18 Groups, Leave with Pay	16	17	1	1	1	8	1	2	4		
(31) Leave Without Pay, Absent, Etc.	9	3				2			1	1	1
(32) Shop Master Roll	487	103	47[illegible]	9	31	192	11	31[illegible]	69	58	15

(Feb. 1[illegible])

AUL AND GENERAL SHOP EXPENSE

71,80	74		Total	Machinery Foreman Shop											
				21	28	41	41,612	88	89	94	95	888	Total	General Total
25	6		671	261	234	35	109	32	85		23	7	786	1457
			85											85
				1	2								3	3
			10											10
								2					2	2
1			11	3	1	1/2	2		1	1/2		2	10	21
14			75		6		1		4				11	86
					9		1						10	10
					4								4	4
11			28											28
					4	4	4		2				14	14
							1						1	1
					1		1						2	2
			2		2	2					1		5	7
			3											3
			4						2		3	3	8	12
			8	2			7						9	17
			3											3
					2								2	2
2	6		62	20	4		1	3	2	65			93	155
1	1		16	6	2	1	8	2	2		1	2	23	38
			22	3			6		6			34	49	71
			4	1		3	3						7	11
3	1		37	15	13	2	6	2	6	2	1	1	48	85
				6	3		7			3			19	19
			53	14	6	2	11	2	10	2	1	4	78	126
1	1		17		6	8			1	1			10	87
58	15		1048	336	897	58	158	43	143	714	30	53	1190	2834

(Sheet 1(a))

Object	Days Work	Allotment Nov.(25 Day)	Allotment 11-27-40	Estimated Total	Total
PARCHEAR	0	9562	16827	17969	16
[illegible]	0	7646	[illegible]75	22468	21
[illegible]	4	11972	15072	16300	9
[illegible]	4	9000	22629	23100	9
[illegible]	4	7900	19499	25000	8
[illegible]	4	4800	13775	15200	1
COLORADO	4	4000	17405	20300	
J.F.[illegible]				13500	
[illegible] YORK				27000	
[illegible]				13500	
		(50720)			
		Index 94.4.			
MISC. SHIPS	4				
Yard Craft and Dredge Repair		1890			
Ships Out of Commission		1890			
Classified		9480			
Other "X"		1480			
Title "Y"		650			
Title "Z"		1600			
Title "C"		330			
Title "D"		500			
Total		68420			
Allotment		78480			
Available		Index 95.7			
Unobligated		4000			
Unobligated					

Form OPD No. 1. Paper Bound Navy Yard. 11-4-40 [illegible]

Nov. Allotment: [illegible]
Tentative Dec. Allotment: [illegible]

[illegible] Department

[illegible] Sheet Letter —[illegible]

Supplement Engg. (601-1) All Files. Date 5135 PM 26 Nov. 1966.

Total Receipts to date	Received this month	Projected Requirements to end of month	Total projected Requirements for this month	Rev. (25 Da) Allocation	Rev. (26 Da) Allocation
16661	4211	0	4211	-	
21566	11468	0	11468	-	
9217	9217	Rev. 12x117 2340	12277	8865	-
9201	9201	Rev. 12x167 4020	13221	9408	-
4994	4994	Rev. 12x159 3510	8504	10885	-
1096	1096	Rev. 12x230 12630	23598	12834	-
61	61	Rev. 12x23 5485	6434	176574	30000
			-	35000	100000
			-	8000	250000
			-	1000	125000
	(40542)				
	245	355	600	495	800
	2983	Rev. 12x51 1290	4273	1890	1965
	1254	146	1600	1890	1965
	7611	1809	9420	9420	9780
	963	457	1420	1420	1475
	115	185	300	600	640
	638	266	900	1600	1600
	245	85	330	330	340
	709	91	800	900	500
	58899	17421	78480	75781	69065
			72480	63480	
		17421			
				12301	

L 72480 I 28080 K 21500 P 122000
63480 30780 14000 100000

(Fig. 2)

Ships	Class Num's.	Allotment Nov.(25 Dec)	Balance 11-27-[illegible]	Recommended Total	Tot
FARQUHAR	8	585	9930	5150	
THOMPSON	8	595	4413	5150	
DECATUR	4	1200	3333	4800	
OUTAMA	4	2800	5990	5800	
MOODY	4	3000	9987	6450	
REILLY	4	1400	1665	3700	
COLORADO	4	2000	4461	7700	
J.F.BURNES				3400	
NEW YORK				10000	
SUMNER				3400	
MISC. SHIPS	4	1800			
Yard Craft and Foreign Ships		500			
Ships Out of Commission		50			
Classified					
Other "G"		1000			
Title "F"		500			
Title "E"		100			
Title "C"		100			
Title "S"		50			
Total		17060			
Allowance		21500			
Available					
Unobligated		4440			
Unobligated					

Form 6078 No 1. Signal Ground Navy Yard 11-[illegible] 1944

Nov.Allotment: 1 78[illegible]

Tent. Dec.Allotment: 6[illegible]

INDUSTRIAL DEPARTMENT

FINANCIAL ANALYSIS [illegible] — Statement

Appropriation Misc. (501-1) All Fields. Date 4:45 PM 24 Nov. 1926.

Sanctioned Total	Total Expend'd to date	Expended this month	Proposed Expenditure to end of month	Total proposed Expenditure for this month	Dec. (25 Da) Obligations	Jan. (26 Da) Obligations
1150	5778	965	0	965		
1150	2869	792	0	792		
1800	1914	Cr.987	1119	132	300	
5800	2087	2087	2903	4990	1000	
5450	7215	7215	1772	8987	1000	
3700	1	1	4998	5008	3200	
7700			1000	1000	6200	500
1400					700	7700
0000					20000	65000
2400					1000	30000
		62	94	156	100	700
		469	331	800	500	500
		71	29	100	100	100
		402	598	1000	1000	1000
		1	199	200	500	500
		136	64	200	100	100
		74	26	100	100	100
		852	48	300	200	200
		11540	8682	20822	17100	15900
				21500	16000	
			9960			
			1278	1278		
					1100	

L 75400 I 28080 M 21500 T 122000
63400 20580 16000 100000

(Fig. 3)

NOVEMBER

FORCE DI

(25 Working Days)	STD.	1	2	3	5	6	7	8	9	10	12	13	14	
	82	86	87	83	86	85	82	83	85	83	86	86	84	8
	11	12	6	3	12	11	4	9	7	3	17	11	3	
	107	35	70	121	80	74	79	60	60	139	86	71	81	
	100	105	112	106	113	108	105	110	110	112	118	111	108	1
	54	71	79	60	40	33	34	61	80	56	49	45	35	5
	112	99	95	74	67	76	69	73	69	73	82	81	89	8
	42	41	40	40	32	37	40	35	40	29	26	34	16	
	312	148	171	148	139	175	157	130	135	147	161	173	172	1
	375	366	307	289	288	302	296	306	289	303	307	313	325	3
	44	52	48	49	48	40	51	54	60	39	57	59	50	4
	33	24	24	39	29	22	29	29	22	40	28	24	16	
	23	21	20	21	20	20	21	21	21	21	20	21	21	
	72	84	84	72	81	84	80	80	84	74	77	75	77	
	1922	1944	1909	1944	1934	1918	1917	1911	1919	1903	1952	1934	1907	1
	—	—	—	—	—	—	—	—	—	—	—	—	—	
	300	229	230	217	251	239	231	239	246	222	222	248	236	2
	50	49	55	51	63	29	39	65	55	62	69	68	75	
SARATOGA - 16A	750	739	750	966	1013	970	982	1001	1004	941	951	947	955	9
LEXINGTON - 16A	—	—	6	12	9	10	8	11	14	3	2	15	15	
TOTAL FORCE	2374	2391	2361	2362	2367	2371	2371	2371	2372	2372	2378	2370	2374	2

RCE DISTRIBUTION

13	14	15	16	17	19	20	21	22	23	24	26	27	28	30	AVE.	TOTAL
66	84	81	89	83	88	88	88	86	86	86	85	83	84	80	85	2117
11	9	14	7	3	19	13	8	17	11	5	13	13	9	8	10	248
71	81	96	97	162	96	69	67	70	74	137	125	122	134	2514	100	25044
111	105	114	104	90	130	114	104	109	98	103	109	110	107	96	108	2707
45	35	57	66	65	65	65	64	70	85	77	79	66	67	63	64	1600
81	89	89	101	70	75	90	101	91	91	109	122	111	125	128	92	2296
34	16	13	3	5	4	15	19	34	21	35	61	64	45	52	31	701
175	172	166	210	220	234	256	261	295	326	344	486	580	527	471	240	6190
313	328	321	324	319	364	362	426	477	523	596	592	610	513	550	385	9626
59	50	40	44	44	48	56	54	57	40	56	51	51	55	49	49	1226
24	16	25	23	32	17	10	24	19	19	37	27	23	15	312	26	6401
21	21	21	21	20	21	21	21	21	21	21	21	21	22	21	21	321
75	77	71	71	71	74	73	69	73	72	67	76	73	76	73	76	1892
1974	1987	1991	1985	1923	1884	1934	1987	1949	1973	1825	1776	1781	1796	1647	1922	48089
—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
240	236	230	231	220	230	251	247	264	269	272	324	313	310	317	252	6319
65	73	92	82	77	79	79	47	38	40	32	46	50	53	42	59	1462
247	253	247	226	214	250	223	241	240	271	291	15	10	5	3	135	18388
15	15	9	0	10	5	5	3	2	10	3	0	6	4	4	7	174
2370	2394	2377	2360	2298	2384	2377	2362	2360	2380	2270	2286	2223	2259	2218	2542	59642

(F.e. 4)

NAVY YARD, PO
20 Nove
Shop 111. ESTIMATE OF DA

Line No.		Labor	Rates of Pay. No. emp Cash Rate
1	01 Misc.Labor & Consumable Supplies	259.36	$4.46 (2) Janitors. L Mon. Dressing Room
2	02 Office	14.00	Printing
3	02 Shop Store	68.16	$7.12 (1); $4.56 (2).
4	02 Tool Room	400.00	Toom Room Employees.
5	03 Fuel		
6	11,12,13,14 & 15 Scrap Credits, Etc.	49.32	Scrap Credit ($115.00
7	17 & 18 Leave & Holiday	4667.00	
8	Supervision	3246.00	(1) 14.00; (2) 10.16; (1) 8.64; (2) 7.68; (
9	A Sub-Total Operating Expense Lines 1 to 8	8705.84	No Leave Figured.
10	3 Buildings, Repairs to		
11	40-41 Misc. Equipt.	40.00	Tool Room Employees.
12	5 Plant Appliances	119.00	Misc.Shops.
13	6 Machinery and Machine Tools	330.00	Same as Above.
14	7 Portable Power Tools	200.00	Tool Makers.
15	8 Loose and Hand Tools	400.00	Tool Makers and Helpe
16	B Sub-Total Maintenance, Lines 10 to 15 Inc.	1089.00	
17	C Sub-Total A & B less Leave & Holiday		
18	D Total Direct Shop Expense A & B	9794.84	
19	Power Transfer		
20	General Expense Transferred		
21	Grand Total		
22	Estimated No.Employees on Productive Labor	370.00	
23	Estimated Average Pay 6.20		
24	Estimated Productive Labor 370 x 6.20 x 23		
25	Estimated Shop Expense Rate 32.32		
26	Estimated S hop Expense Rate, exclusive of leave and holiday 13.3		

F

T YARD, PUGET SOUND, WASH.
20 November 1928. (23 Working Days).
IMATE OF DAILY SHOP EXPENSE. MONTH OF DECEMBER 1928.

Dy. No. Employees At: Cash Rate	Material	Brief Description of Materials	Total	Remarks
Janitors. Look Up ing Emery Wheels.	366.00	Lub.Oil Waste,Lamps,Batteries, Hack Saw Blades, Chalk, etc.	645.36	
			14.00	
$4.88 (2)			68.16	
Employees			400.00	
	125.00	Fuel Oil and Coke	125.00	
			49.32	Time Allowed and Instruction of App.
			4667.00	
(2) 10.16;(1) 10.56; (2) 7.68;(6) 8.72			3748.00	
figured	513.00		9218.84	
		Less-Scrap Credit	115.00	
			9103.84	
Employees	20.00	Elec.Cables,Plugs,Sockets,Etc.	60.00	
	75.00	Materials used in Repairing Motors and Crane Mach.	194.00	
ers	180.00	Tool Steel, Mild Steel, Cast Iron, Brass, Etc	510.00	Includes $100.00 Fire Brick for Angle Furnace
s	200.00	Valves,Bearings,Bushings, Air Hose, Pneumatic Mach.	400.00	
s and Helpers,Gen.	900.00	Tool Steel,Drills,C.S.K.Rivet Sets,Reamers,Emery Wheels,Tool Box Equipment, Etc.	1300.00	
	1375.00		2464.00	
			7015.84	
	1888.00		11682.84	
			2114.00	
			3189.00	
			16980.84	
			1853.00	
			585.32.00	
13.]				

(Fig. 5)

COMMENT.

By Captain S. M. Robinson, U. S. N., Member.*

The Industrial Department of every Navy Yard, on the first day of the month, finds itself faced with the following situation:

(1) It is given a certain specified sum of money to be spent.

(2) It has a definite date of completion for each vessel at the Navy Yard.

Since the money allocated for the repair of ships is never entirely adequate, it is desirable — in fact necessary — to spend each month, practically the total sum of money allocated but at the same time this sum must not be exceeded. Combining this condition with the necessity for the completion of ships on certain specified dates makes the problem become quite complex, and it is very obvious that some means must be devised for looking quite accurately into the future. Otherwise there will be an excessive labor turnover; work will be done inefficiently by excessive use of shifts; or else a great deal of necessary work on ships will not be done.

The methods laid down by the author of this paper have been in operation at this Navy Yard for over two years and have been very successful.

There is one other point in this article that should be stressed and that is the budgeting of shops. This is one of the primary causes which have resulted in reducing the overhead at this Navy Yard from 55 per cent to 35 per cent in a little over one year.

* Industrial Manager, U. S. Navy Yard, Puget Sound, Washington.

THE HIGH SPEED OIL ENGINE.*

By E. C. Magdeburger.†

The engineering science of today points to the ideal high speed oil engine of the future as the engine which will develop the most power out of a given weight of materials used and will do so with the least expenditure of fuel. To make headway towards such an ideal requires the highest expression of man's genius in conquering the forces of nature. The goal, however, is so enticing that many an enthusiastic engineer follows the procession which some day will arrive at this ideal. Meanwhile, we are content to review their achievements in the hope that new recruits will catch the glimpse of this the most interesting and challenging problem of humanity today, and that greater steps toward success may follow. It is with this ideal in mind that important contributions toward the solution of the problem will be reviewed, without going into the more common details of the engines considered. The terms oil engine and Diesel engine will be used indiscriminately to denote the type of engine using the heat of compression for ignition.

The principal advantages of any Diesel engine are (1) its high thermodynamic efficiency which can be best expressed in its ability to develop one brake horsepower out of as low as .36 pound of fuel, or converting 30 to 35 per cent of its thermal energy into mechanical energy; (2) its ability to burn fuels of higher specific gravity and lower cost; (3) its instant readiness to start and develop full rated power within a few minutes, which it possesses in common with the gasoline motor; (4) the same low fuel consumption may be obtained in a cylinder of one horsepower as well in a giant of 2500 horsepower; (5) a flat fuel consumption curve, the consumption varying but little with the changes in load between full load and say one-half load.

* Presented before the Buffalo Section of the Society of Automotive Engineers, 11 May, 1929.

† Aide on Diesel Engines, Navy Department.

When compared with gasoline engines, its advantages further include (1) the use of non-inflammable fuel and therefore safety of operation, and (2) the omission of spark plugs, magnetos and other electrical appliances required for ignition, including the batteries, unless these are used for starting the engine by means of an electric motor.

The main reason for the high speed type of oil engine is the same as for any high speed machinery — the saving in space and weight through increased rotative speed.

If saving in weight and space is the principal objective of the use of high rotative speed, other means which accomplish the same end would naturally be incorporated in the design of high speed oil engines as well as high speed. The number of such means used and the degree of their refinement determine the rung of the ladder to which any given effort belongs and its proximity to the ideal above mentioned. As yardsticks will serve the fuel consumption and the weight per brake horsepower which together with other pertinent data shall be tabulated for quick comparison. It must be noted here that due to lack of uniformity of test conditions for obtaining fuel consumption data some manufacturers have refused permission to publish any — the available figures are published without comment or any reference to the conditions under which they were obtained. Direct comparison of fuel consumption figures is therefore made rather difficult.

Most prospective applications of a high speed oil engine require also that it be " flexible," i. e., that it would run smoothly at a very low speed while idling and at any speed between idling and full power under any load. This is quite a problem to solve, and the ingenuity of the many solutions it is also proposed to review.

AIR INJECTION ENGINES.

Historically the *Maybach* four-cycle oil engine (see Figure 1, also line 17, Table I) is perhaps the first in which heavy oil burning has been applied to a high speed type of gasoline engine. Since its introduction in 1923 it has only been built in one size, developing a maximum of 150 B.H.P. at 1300 R.P.M., with a fuel consumption of .40 —.49 pounds per B.H.P. per hour. This engine has many interesting mechanical features, such as the use of roller

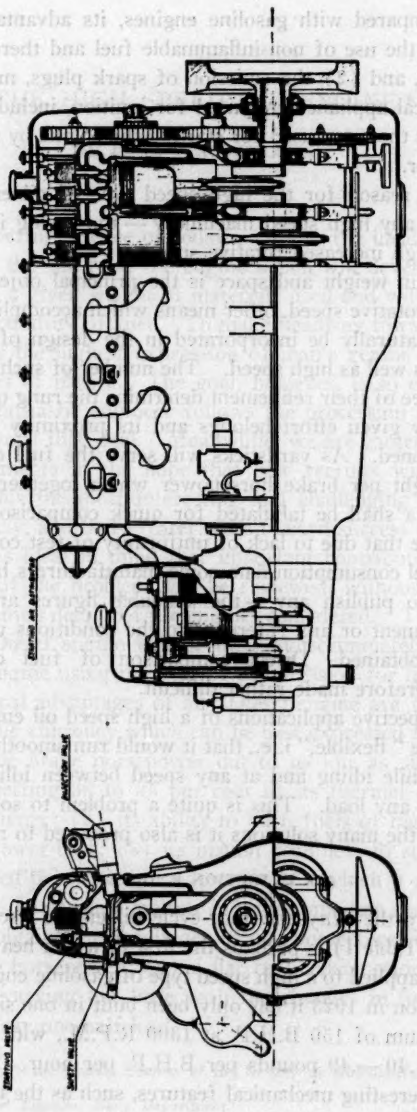


FIGURE 1.—MAYBACH ENGINE.

bearings in the seven main bearings and the crank pin ends of connecting rods, air intake through the crank case, for deadening the intake noise and ventilation of the crank case, but fuel injection by means of high pressure air, compressed in a separate three crank, three stage air compressor, has been retained from the conventional type of Diesel engine. Compression pressure of 550 pounds per square inch has been adopted, while injection air pressure up to 1500 pounds per square inch and starting air pressure of 800 pounds per square inch are required.

Its fuel valves have a constant lift, but unique means for automatically changing the cross-section of the nozzle in accordance with the load have been developed, and a pressure regulator is provided to regulate the injection air pressure to suit the engine speed. This makes the engine adjustable to a wide range of torque and speed, and capable of operation under remote control or bridge control. Particular attention to balancing of all rotating and reciprocating parts, and the use of a torsional vibration damper on the forward end of the engine are safeguards against disagreeable and often dangerous vibration, which made the high speed of the engine possible. Its fuel pump is a unique structure of vertical plungers pressed by springs against a common "wabble plate" driven by a vertical shaft. The change of inclination of this plate changes the effective stroke of the pump plungers.

A large number of these engines are reported to be installed in railcars operating on German, Danish, Swedish and Hungarian railroad systems where such rigid schedules as eighteen hours out of every twenty-four are being maintained.

An even older successful attempt to use heavy oil in high speed engines must be attributed to Hindl, one of the original collaborators of Diesel, who built a number of motorboat engines during the war in Austria. His system of fuel injection by means of hot high pressure air has been applied to a two-cycle engine developed by Norman McCarty for the Climax Engineering Co., of Clinton, Iowa. The essential features of this engine are shown in the cross-sections, Figure 2, and the data given in line 8 of Table I. A small amount of air at 215 pounds pressure is delivered by an independent air compressor to the suction line A of the engine injection air compressor, of which there is one for each working

cylinder, driven from the common camshaft of the engine. Here the air is compressed to a pressure of about 1100 pounds per square inch and, in highly heated condition, forced through the fuel valve and into the cylinder, the pressure rising up to about 1300 pounds per square inch at the end of injection. A fuel consumption of .47 pound per B.H.P. hour is claimed [1] for this engine. A special

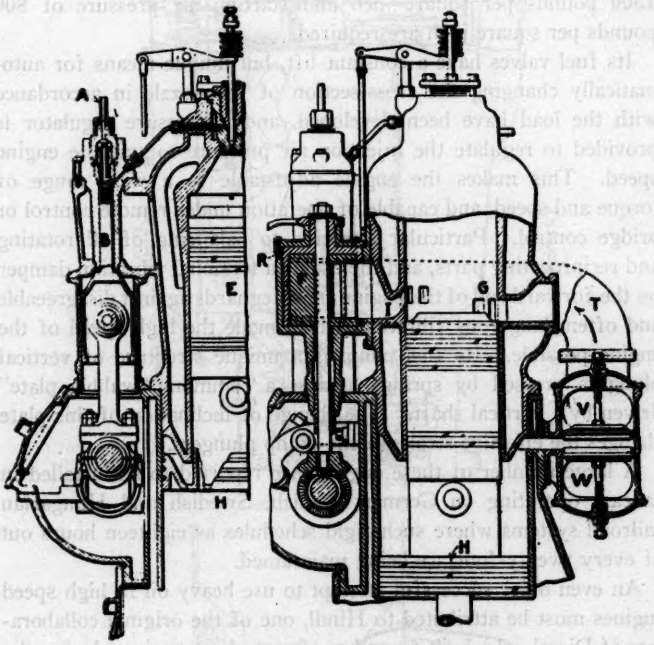


FIGURE 2.—CLIMAX-MCCARTY ENGINE.

feature of this engine is the mechanically operated rotary valve (F) controlling scavenging and supercharging of the working cylinder. It will be noted that the scavenging ports (I) remain open after the piston on the upward stroke has closed the exhaust ports (G) and supercharging of the cylinder up to the scavenging receiver pressure becomes possible. Furthermore valve control of scaveng-

[1] Description of engine, by Norman McCarty—Power, March 12, 1929, p. 429.

ing permits the adaptation of time available for scavenging to the speed of the engine so that no useless waste of air need occur at slow speeds when the time the ports remain open is longer and more air would flow through them and out into the exhaust until the compression pressure might become too low for reliable ignition. Whether the use of stepped piston for furnishing scavenging air contributes to the reliability of the engine is a debatable question. Suffice it to say that it was abandoned by a number of engine builders and the experience of some early submarine engines [2] using this construction has been unsatisfactory.

AIRLESS INJECTION OF FUEL.

The most obvious means of saving weight and space is of course the doing away with the injection air compressor and to this end much experimental work was devoted ever since 1910, when McKechnie of Vickers, Ltd., took it up seriously. Dr. Diesel, himself, of course, had tried it, but soon discovered in the high pressure air the ideal agency for storing the energy required to break up the oil fuel into suitable droplets while passing through the fuel valve when the needle is lifted (atomization), to carry these droplets the farthest required distance from the valve (penetration) and by creating the inevitable turbulence within the combustion space to bring the fuel droplets into the intimate contact with the oxygen required for the reaction of combustion. All of this is necessary if maximum M.I.P. is to be developed within the cylinder and efficient combustion with lowest fuel consumption obtained.

To produce all of the above phenomena without the heavy and often troublesome air compressor is quite a problem, and two distinct types of injection have been developed, (*a*) fluid injection, which relies upon the pressure of the oil issuing from the injection valve for atomization and penetration while using other means to promote turbulence of the air charge and efficient combustion, and (*b*) gas injection which depends upon gases of partial combustion in a precombustion chamber for atomization and penetration of the main fuel charge, and may or may not use other means to create turbulence of the air charge. The fluid injection in turn can be

[2] Diesel Engines in Submarines, by E. C. Magdeburger, JOURNAL OF AMERICAN SOCIETY OF NAVAL ENGINEERS, Aug. 1925, p. 572.

subdivided into two different methods, namely, (*a*) constant pressure fluid injection, which depends upon fuel maintained under constant pressure of 2000 to 8000 pounds per square inch, in a manifold, from which the individual cylinders receive their charges by means of mechanically operated needle valves, the latter doing the metering of the charges, and (*b*) variable pressure or direct pump fluid injection, where the pump does both the metering and the injection, either through a spring loaded needle valve on the cylinder or through an open passage leading into the cylinder. Each of these methods has, of course, its advantages and disadvantages and therefore its partisans. Before proceeding further with the description of individual systems of airless injection the disadvantages of each as viewed by its competitor will be cited so that later these can be checked and the degree of their probable importance estimated.

Constant pressure fluid injection has the following disadvantages:—(*a*) with fuel under constant high pressure in the manifold, the slightest leak in the injection valve will cause dripping which will cause smoky exhaust, deposit coke on the piston and gum up the piston rings; (*b*) metering minute quantities of fuel by keeping the injection valve open for a definite length of time is hardly reliable, since it fails to account for changes in the orifice of the valve due to erosion and clogging; (*c*) the injection control by the lift of the valve is extremely sensitive, hence continuous changing of the adjustment of individual cylinders becomes necessary.

The direct pump system employs a pump to do all the metering and changing of the timing of the fuel injection. The metered charge passes at the proper time through the discharge valve of the fuel pump and in many cases directly into the cylinders of the engine or through a spring loaded check valve on the cylinder. Among the troubles to guard against are (*a*) after dripping due to secondary discharge from pressure waves and air pockets in the line or sluggishness of valve seating; (*b*) misfiring due to the presence of air in fuel; (*c*) variation of injection lag from compressibility of oil and breathing of the pipes leading from fuel pump to cylinder.

The precombustion chamber injection consists essentially of a chamber, cup or cavity connected either with a number of small holes or one large one with the combustion space in the cylinder, and into this chamber the fuel is injected by a pump through a spring loaded needle valve. The gases resulting from combustion of a small part of the charge supply the necessary energy for atomizing and distributing the bulk of the fuel charge through the main combustion space in the cylinder. With the injection chamber method, as viewed by advocates of other systems of injection such as Dr. W. Riehm of the M. A. N. Co., "the timing of the injection requires much more serious consideration than in pressure atmomization." He further states [3] that a spring loaded valve "involving a number of delicate parts is an inconvenience in manufacture and does not promise reliability of operation. Especially the nozzle tip, in which are the valve seat and the ports, and the needle valve are subject to great wear." Of course, part of the above criticism applies to any but the open nozzle type of the fuel valve employed in M. A. N. engines.

With these criticisms for a guide, engines employing the various airless fuel injection methods will be reviewed.

CONSTANT PRESSURE FLUID INJECTION.

This system was originally introduced by McKechnie of Vickers, Ltd., on submarine engines of their manufacture, and of late years adapted by a number of Diesel engine builders in this country and used also on their high speed models. It consists essentially of a "common rail" or a fuel manifold containing fuel under pressure of from 2000 to 8000 pounds per square inch, a constant stroke fuel pump and variable lift injectors mounted on each working cylinder. The pressure in the system is maintained by one or more accumulators which often are round steel flasks, the elastic walls of which prevent pressure fluctuation. The fuel pump which for the sake of uniformity of pressure often consists of three individual plungers timed 120 degrees apart delivers a constant amount of fuel into the pressure manifold from where the excess of fuel is returned to the suction side of the pump by a relief valve set at a

[3] High Speed Automotive Diesel Engines, by Dr. W. Riehm, S. A. E. Journal, May, 1928, p. 526.

given pressure. The metering of the charge is done by mechanically operated injection valves, controlled by the governor or a throttle or by both of them. Throttling reduces the time of injection and the pressure maintained in the manifold simultaneously.

A number of manufacturers of high speed oil engines in this country have adopted this system for the sake of its simplicity. The oldest high speed engine using this type of fuel injection is the Atlas-Imperial engine (Figure 3), which has been used extensively by a prominent manufacturer of excavators. It presents nothing spectacular in its design, but has a proud record of seven years of successful achievement to boast of. Line 1, Table I, gives the essential details of this type of engine.

The Winton Company also uses this type of fuel injection in their high speed models, although the older standard line of air injection engines is still manufactured. The dimensions of three engines of this type are given in line 23, Table I. Twin valves are used for both inlet and exhaust, with the fuel valve in the center.

The Consolidated Shipbuilding Corporation has recently placed on the market a high speed "Speedway" engine, using also this type of fuel injection system. Line 19, Table I, gives some of the data pertaining to this engine.

The Standard Motors Construction Company builds a smaller cylinder also with this type of fuel injection, the details of which are given in line 20 of Table I.

The Bessemer Engine Company has built one high speed engine for railway service using 8 cylinders arranged in fours on each side of V, and is manufacturing a standard line of engines the cross section of which is shown on Figure 4, also showing the little wedge used to vary the lift of the valve for metering purposes. Line 5 of Table I gives the numerical data and the type of spring loaded valve developed by this company is shown on Figure 5.

The most courageous attempt to reduce the weight of a larger sized engine by a special design using special materials, V type of construction and constant pressure type of fuel injection was made by Treiber [4], [5] who designed a 3000 horsepower engine to weigh

[4] High Speed Diesel Engines, by O. D. Treiber, S. A. E. Journal, February, 1928, p. 183.

[5] Modern Trend of the Diesel Engine, by O. D. Treiber, Trans. Soc. of Naval Architects and Marine Engineers, 1927, p. 235.

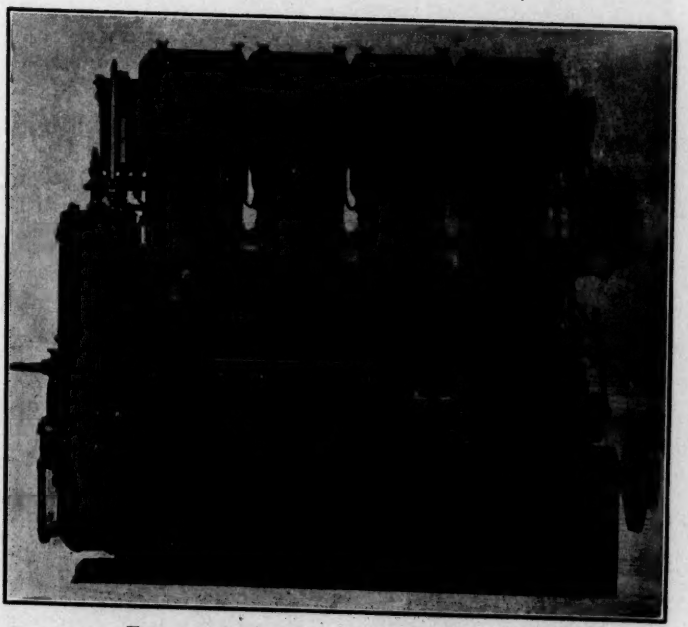


FIGURE 3.—ATLAS-IMPERIAL ENGINE (7 × 8½).

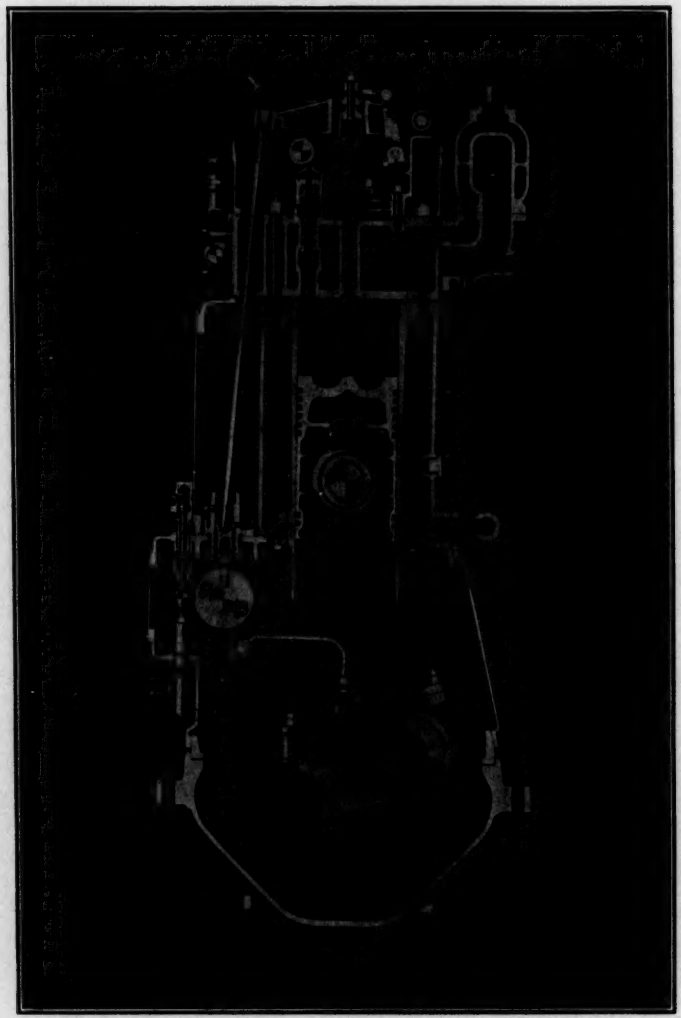


FIGURE 4.—BESSEMER ENGINE (7½ × 10½).

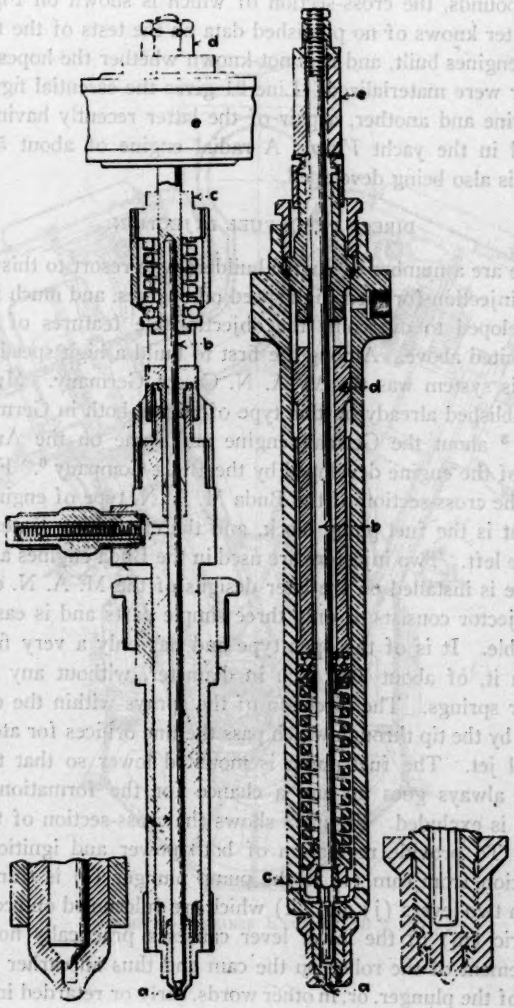


FIGURE 5.—FUEL VALVE TYPES OF BESSEMER ENGINES.

63,900 pounds, the cross-section of which is shown on Figure 6. The writer knows of no published data on the tests of the first set of two engines built, and it is not known whether the hopes of the designer were materialized. Line 21 gives the essential figures of this engine and another, a pair of the latter recently having been installed in the yacht *Vara*. A radial engine of about 500-600 B.H.P. is also being developed.

DIRECT PUMP FUEL INJECTION.

There are a number of engine builders who resort to this system of fuel injection for their high speed oil engines, and much ingenuity developed to overcome the objectionable features of such a system cited above. Among the first to build a high speed engine with this system was the M. A. N. Co. of Germany. Much has been published already on this type of engine both in German and English [3] about the German engine and some on the American design of the engine developed by the Buda Company [6]. Figure 7 shows the cross-section of the Buda M. A. N. type of engine. On the right is the fuel pump block, and the circulating water pump is on the left. Two injectors are used in the Buda engines although only one is installed on the later designs of the M. A. N. engines. This injector consists of only three simple parts and is easily dismountable. It is of the open type and has only a very fine hole through it, of about .006 inch in diameter, without any moving parts or springs. The direction of the sprays within the cylinder is done by the tip through which pass the fine orifices for atomizing the fuel jet. The fuel pump is mounted lower so that the fuel column always goes up and a chance for the formation of air pockets is excluded. Figure 8 shows the cross-section of the fuel pump. To provide regulation of both power and ignition time, the motion from cam (i) to the pump plunger (l) is transmitted through two levers (j) and (k) which are fulcrumed on eccentrics. Eccentric (m) of the lower lever causes a practically horizontal displacement of the roller on the cam and thus an earlier or later stroke of the plunger, or, in other words, early or retarded injection. Eccentric (n) of the upper lever is shifted in an almost vertical

[6] The Automotive Full-Diesel Engine, by R. J. Broege, S. A. E. Journal, February, 1928, p. 177.

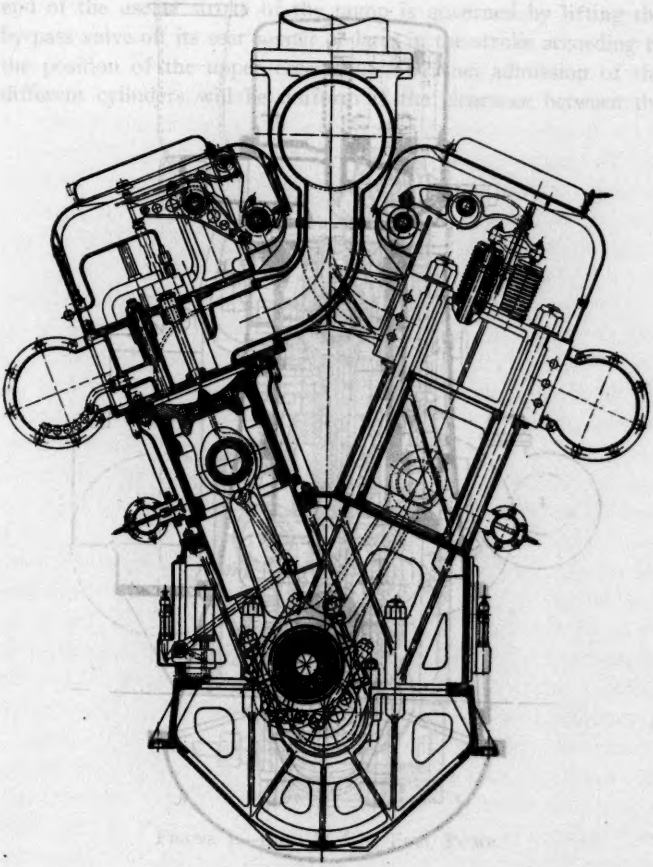


FIGURE 6.—TREIBER ENGINE (16 × 16).

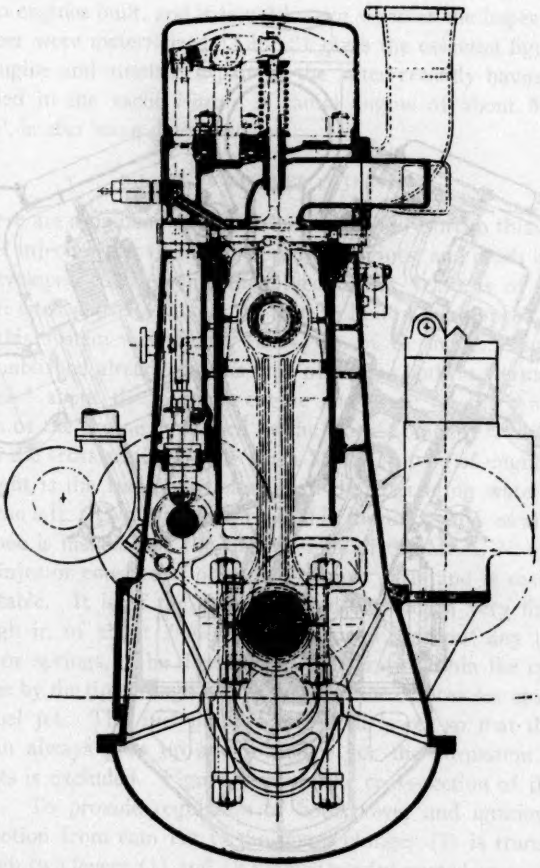


FIGURE 7.—BUDA-M.A.N. ENGINE (6″ × 8″).

direction and thus changes the clearance between the set screw (p) in the lever and the by-pass valve (q) in the pump body. The end of the useful stroke of the pump is governed by lifting the by-pass valve off its seat sooner or later in the stroke according to the position of the upper eccentric. The fuel admission of the different cylinders will be uniform if the clearance between the

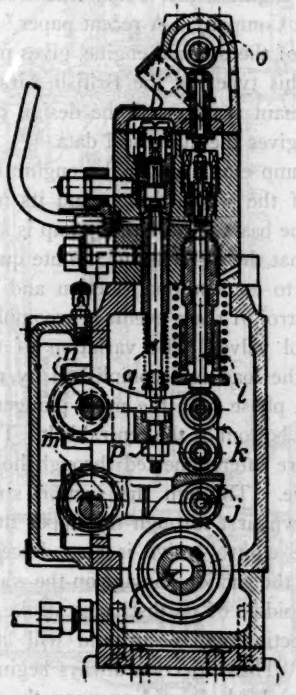


FIGURE 8.—BUDA-M.A.N. FUEL PUMP.

overflow valve and the set screw is the same, in the same relative crank position, for all cylinders. In the two spray arrangement the check valve is located in the fitting where the pipes branch off to the two nozzles. Shaft (o) above cuts out one or more cylinders by depressing the suction valve of the corresponding fuel plunger. Thus the operator can determine quickly by sound whether all

cylinders are equally loaded. Line 7 in Table I gives particulars of two engine sizes manufactured by the Buda Company. A large number of these found their way into railcars, trucks, excavators, motor boats and industrial installations, both in this country and in Europe.

Another remarkable development with this type of fuel injection is the Beardmore engine which is manufactured in this country by the Westinghouse Company. A recent paper [7] by A. E. L. Chorlton, the designer of the original engine, gives pictures of the latest development of this type for the British Airship *R-1*, Figure 9, from which the main features of the design can be studied, and line 3 of Table I gives the numerical data.

The fuel oil pump employed in the engine is the most characteristic feature of the whole design and its main feature is the control valve. The basic idea of this pump is shown schematically on Figure 10 is that the metering of minute quantities of fuel can not be entrusted to conventional suction and discharge valves—hence positive control of these events is accomplished by a separate plunger or control valve. The variation of the fuel charge in accordance with the load is accomplished by revolving the valve which operates in phase with the main plunger and has a helical cut off edge on the bottom of the central belt. The upper and lower annular cavities are interconnected through holes into the hollow body of the valve. Through the suction stroke of the pump plunger oil is drawn first through the upper then from the lower annulus while the control valve is also descending. When the central belt blocks the suction passage on the way up while the fuel plunger is at the middle of its discharge stroke, and its velocity is at maximum, injection will occur and will last as long as the obstruction exists. Thus injection always begins at the same time and varies with the load, depending upon the part of the helical edge, which is turned toward the suction passage, and this is controlled by the governor. The fuel pump is coupled to the engine shaft and each plunger is made to serve two working cylinders through a switch valve operated at half engine speed and connecting either one or the other of the two cylinders served, as shown on

[7] Oil Engines for Aircraft and Railways, by A. E. L. Chorlton. Engineering, September, 21, 1928, p. 375.

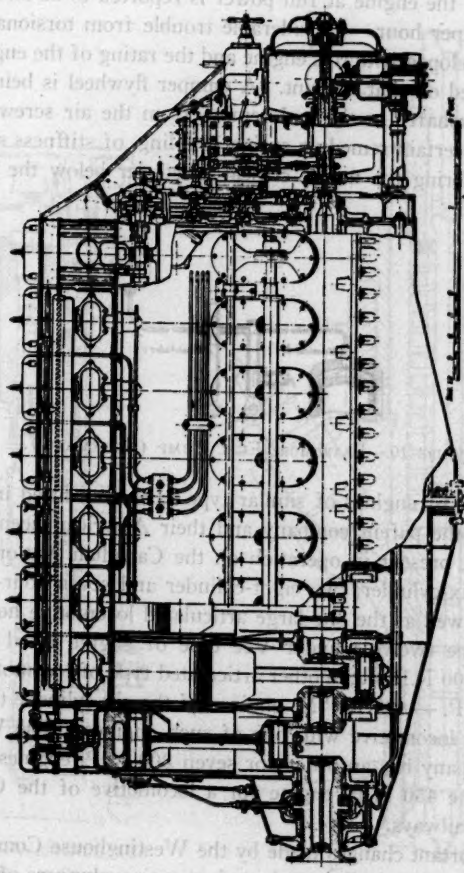


FIGURE 9.—BEARDMORE ENGINE FOR AIRSHIP R-101.

Figure 11, where A points to the rack rotating the control valve. The type of injection valve is shown on Figure 12. The fuel consumption of the engine at full power is reported to be .385 pound per B.H.P. per hour. Considerable trouble from torsional vibration has developed with this engine and the rating of the engine had to be reduced on that account. A damper flywheel is being fitted to the crankshaft [8] at the end remote from the air screw and as a still more certain remedy a spring coupling, of stiffness so calculated as to bring the major critical speed far below the running range.

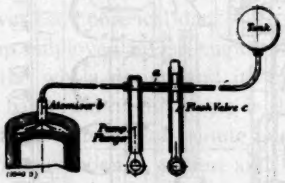


FIGURE 10.—BEARDMORE FUEL PUMP (SCHEMATIC).

A number of engines of similar type have been fitted into locomotives by the parent company and their American licensee, and there are at present in operation on the Canadian National Railways five six-cylinder, two eight-cylinder and seven four-cylinder engines, as well as the one large articulated locomotive housing in each cab one twelve-cylinder Vee type of engine rated at 1340 B.H.P. at 800 R.P.M.; another articulated type of locomotive with two 300 H.P. — 800 R.P.M. engines of the six-cylinder type, and a switching locomotive with one of such engines. The Westinghouse Company has an order for seven 300 H.P. engines for rail cars and one 450 H.P. engine for a locomotive of the Canadian National Railways.[9]

The important changes made by the Westinghouse Company include the reduction in the number of operating plungers of the fuel pump from nine to six for a six-cylinder engine. This was done by running the fuel pump at one-half of engine speed and the elimination of the switch or transfer valve. Thus each power cylinder

[8] The Machinery Installation of Airship R-101, by T. P. Cave-Brown-Cave. The Journal, The Royal Aeronaut. Soc., March, 1929, p. 179.

[9] Paper before S. A. E., by D. W. R. Morgan, S. A. E. Journal, April, 1929, p. 367.

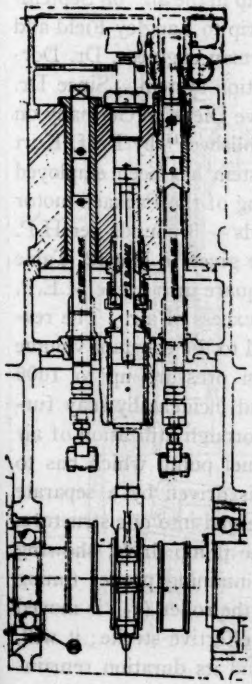


FIGURE 11.—WESTINGHOUSE FUEL PUMP IN SECTION.

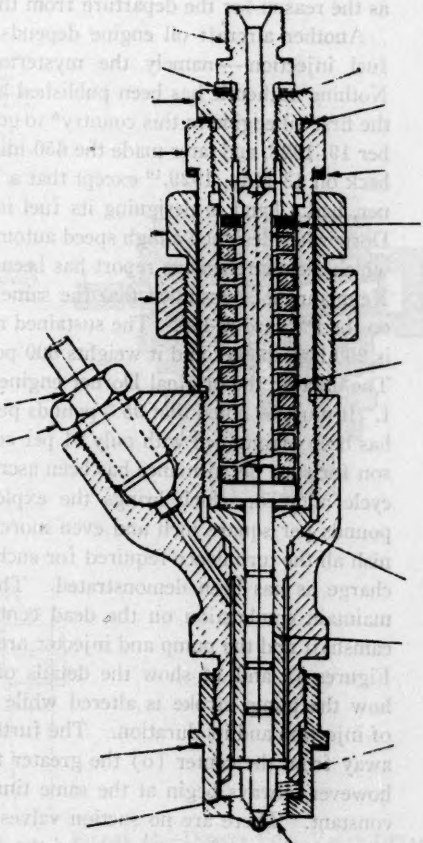


FIGURE 12.—WESTINGHOUSE FUEL VALVE.

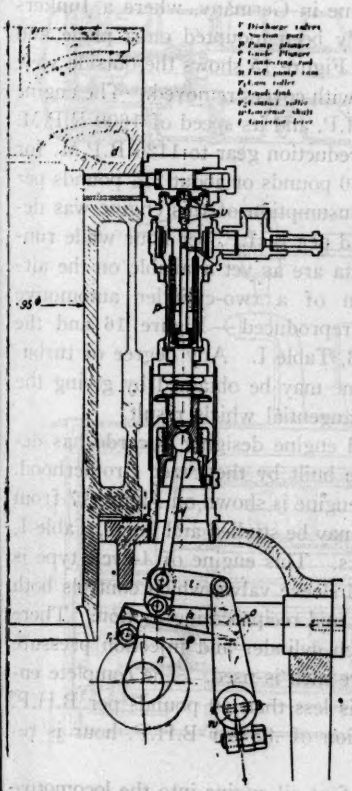
is now provided with its own fuel pump pressure and metering device. Proper distribution of the load between the engine cylinders, which is extremely important in high speed engines, is given as the reason for the departure from the original design.

Another aircraft oil engine depends upon the same method of fuel injection — namely the mysterious Packard radial Diesel. Nothing authentic has been published about this engine which was the first oil engine in this country* to go " up in the air" on September 19, 1928, and later made the 650-mile trip to Langley Field and back on 13 May, 1929,[10] except that a German engineer, Dr. Dorner, had a hand in designing its fuel injection system. Since Dr. Dorner has designed a high speed automotive Diesel in Germany on which a comprehensive report has been published [11] by Prof. Kurt Neumann, it is assumed that the same system has been employed on the Packard radial. The sustained rating of the Packard motor is 200 horsepower and it weights 600 pounds — 3 pounds per H.P. The data on the original Dorner engine are given in line 12, Table I. It must be noted that 95.5 pounds per square inch brake M.E.P. has been maintained with only 21 per cent excess of air. The reason for such performance has been ascribed to the constant volume cycle adopted, which brings the explosion pressure up to 1000 pounds per square inch and even more, and incidentally may furnish all the turbulence required for such thorough utilization of air charge as has been demonstrated. The fuel pump which has to maintain combustion on the dead center is driven by a separate camshaft and the pump and injector are merged into one structure. Figures 13 and 14 show the details of the pump drive, showing how the pump stroke is altered while maintaining proper timing of injection and its duration. The further the roller (r_2) is moved away from the center (o) the greater the effective stroke; it will, however, always begin at the same time and its duration remains constant. There are no suction valves on the pump; the plunger uncovers the suction port (s) and the fuel is drawn in by the vacuum created. Two hardened steel balls serve as discharge valves, and

[10] Packard Diesel—Oil Engine Power, June, 1929, p. 328.

[11] Test of Dorner Automotive Engine, by Prof. Kurt Neumann, V. D. I. 28, May, 1927, p. 775; in English N. A. C. A. Technical Memorandum No. 467.

* An oil engine designed by Bagnulo has been reported as used in training planes by the Italian Government—JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, 1924, p. 523.



IGURE 13.—DORNER FUEL PUMP.

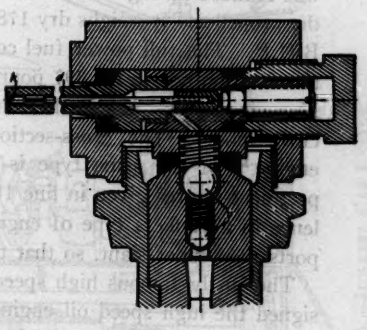


FIGURE 14.—DORNER FUEL VALVE DETAIL.

the fuel, passing them, enters immediately into the injector, which is in the shape of a light poppet valve and by being extremely light follows the movements of the plunger accurately.

In the beginning of February of this year, another high speed oil engine took the air — this time in Germany, where a Junkers opposed piston engine has finally been mounted on a plane and made several successful flights. Figure 15 shows the outside view of the 6-cylinder engine in place with covers removed. The engine develops a maximum of 700 B.H.P. and its speed of 1600 R.P.M. was reduced through a 1 to 1.4 reduction gear to 1120 R.P.M. for the propeller. It weighs dry 1780 pounds or about 2½ pounds per B.H.P. The full power fuel consumption of this engine was determined on test to be .397 pound per B.H.P. per hour while running on gas oil. No further data are as yet available on the aircraft model. The cross-section of a two-cylinder automotive engine [12] of the same type is reproduced — Figure 16 and the pertinent figures given in line 16, Table I. Any degree of turbulence in a Junkers type of engine may be obtained by giving the ports the proper slant, so that tangential whirls result.

The justly famous high speed engine designer Ricardo has designed the high speed oil engine built by the Peter Brotherhood, Ltd. [13] A cross-section of this engine is shown on Figure 17 from which the constructional details may be studied and line 6, Table I, gives its numerical characteristics. This engine of 4-cycle type is the first oil engine to use a steel sleeve valve, which controls both inlet and exhaust by its rotating and reciprocating motion. There is a separate fuel pump for each cylinder and injection pressure of about 5000 pounds per square inch is used. The complete engine weight including flywheel is less than 40 pounds per B.H.P. in cast iron. A fuel consumption of .38 per B.H.P. hour is reported.

The honor of having put the first oil engine into the locomotive cab in this country belongs to the Ingersoll-Rand Company, who, since October, 1925, to date, have placed in service and are building a total of 35 locomotives of four different sizes, namely, 24 of 300

[12] The Junkers Car Engine, by Dr. A. Nagel—Oil Engine Power, April, 1928, p. 263.
[13] The Motorship (British) February, 1929, p. 430.

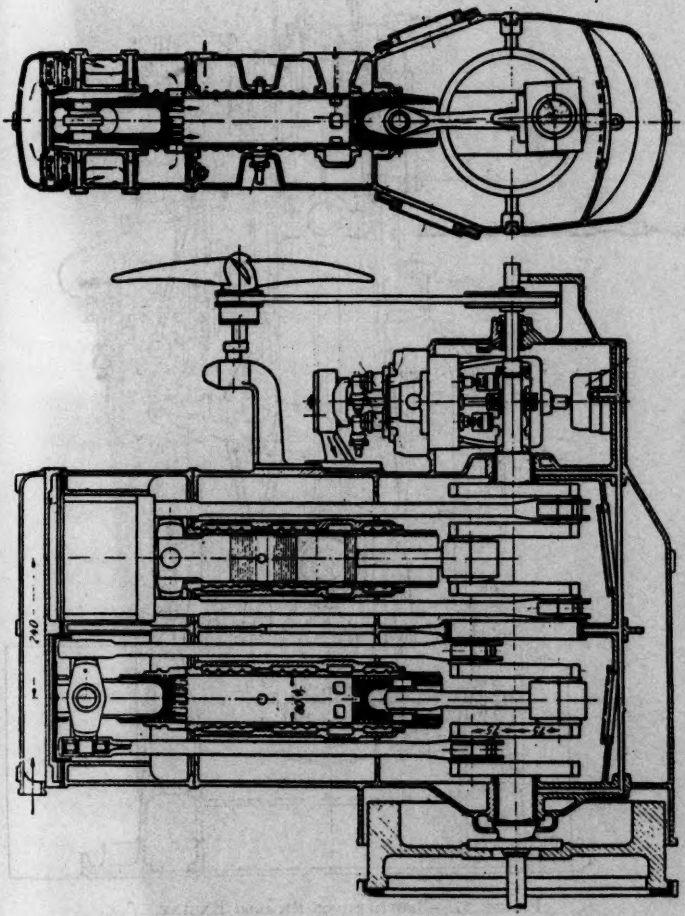


FIGURE 16.—JUNKERS CAR ENGINE.

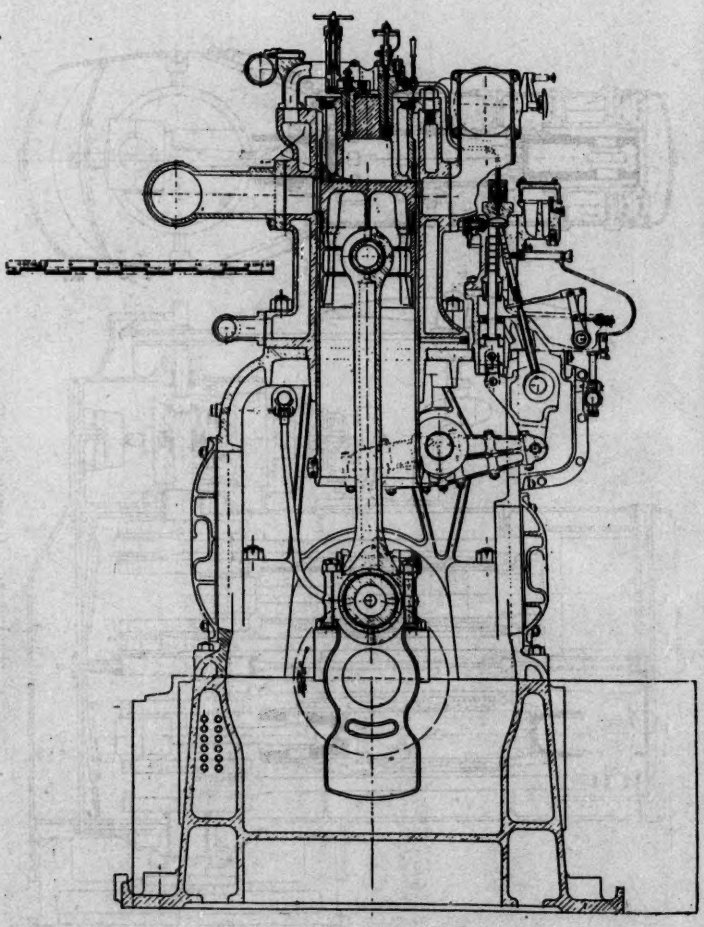


FIGURE 17.—BROTHERHOOD-RICARDO ENGINE.

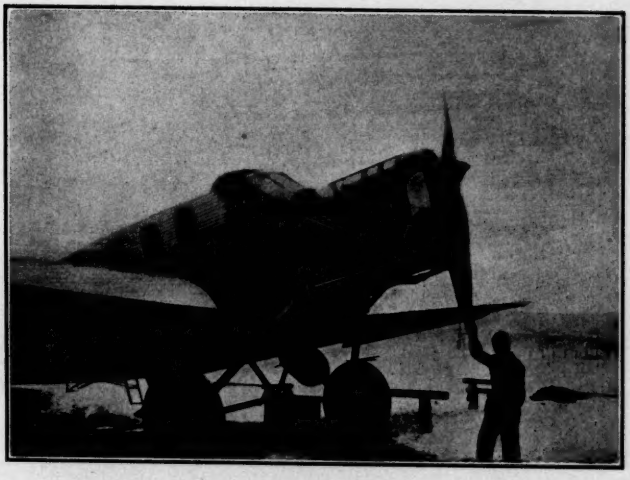


FIGURE 15.—JUNKERS AIRPLANE ENGINE—MOUNTED.

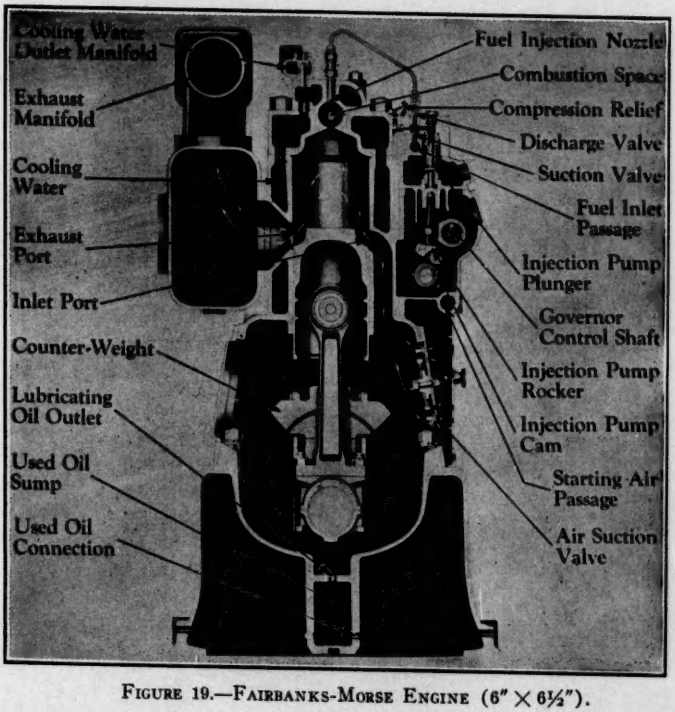


FIGURE 19.—FAIRBANKS-MORSE ENGINE (6" × 6½").

B.H.P., 9 of 600 B.H.P., 1 of 750 B.H.P., and are building one 825 B.H.P. unit. Two different engines have been developed — both of 4-cycle type with six working cylinders. One develops 300 B.H.P. and the other 825 B.H.P. and both using the well known Price system of direct pump fuel injection. This system is based upon impinging of the two diametrically opposite arranged sprays in the center (see Figure 18), and the design of combustion chamber embodies a neck through which the air charge flows at high speed and creates all the required turbulence for efficient combustion of the fuel charge.

Another distinguishing feature of this engine is its fuel system, employing one single plunger pump driven at three times the engine speed for a six-cylinder engine and supplying all six cylinders through a fuel oil distributor. The plunger of the fuel pump on the latter part of its downward stroke pulls open the suction valve which it engages with an open hook in both of these parts, while a spring keeps these hooks engaged. After the suction valve is seated, the spring is being compressed and discharge occurs, which passes through a discharge valve and is controlled by a by-pass valve operated by the governor and pneumatically for quick shut down. The timing of beginning of injection is not affected by the operation of by-pass valve, only the end, and thereby the quantity of fuel injected.

The fuel oil distributor is driven by bevel gears directly from the cam shaft and consists of an intermittent gear which controls the motion of a circular plug with a single hole through it in a chamber connected with the discharge of the fuel pump. In the bottom of this chamber there are six equally distributed holes with connections to the respective cylinder injectors. A spring keeps the distributing plug against this lower face, augmented by fuel pressure during pump discharge. The intermittent gear moves the distributing plug in jerks and only during the suction stroke of the pump, leaving it with its holes matched to the desired hole in the lower face while discharge to the respective cylinder takes place.

PRE-COMBUSTION CHAMBER INJECTION METHOD.

If the size of the output in horsepower were a reliable index of the advantages of the fuel injection system, the Fairbanks-Morse Company, which utilizes this type of injection on their standard

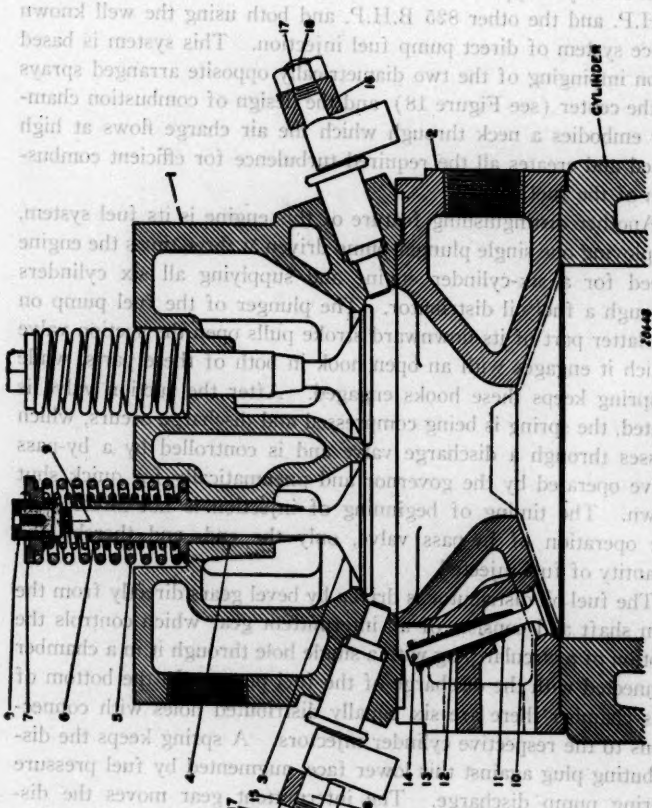


FIGURE 18.—COMBUSTION SPACE OF INGERSOLL-RAND ENGINE.

line of engines as well as on their two high speed models, would no doubt carry away the prize. But a practical solution of any problem is nothing but a compromise and since very much stress in this particular design is laid upon simplicity the pre-combustion chamber method of ignition was chosen. A glance at the cross-section, Figure 19, of this exceedingly compact little unit is convincing proof of that. It is self-contained since it utilizes the crank case for scavenging, although this may be a dear price to pay for simplicity at high engine speed. The fuel injection valve is but a spring loaded check valve and the fuel pump is of variable stroke type, and both simple and compact. Line 13, Table I, gives further particulars of one of the sizes manufactured, although another larger size of cylinder is also manufactured. Among the many applications, such as driving electric generators, pumps, industrial purposes, oil drilling rigs, etc., it is also extensively used in small industrial locomotives.

The original work of Brons and Hvid in this country, who were the first to substitute gases of partial combustion for air for fuel injection, has led to the development of several modifications of their system of fuel injection which have prospered especially in Germany where several different variations are built and where the patent literature on the subject fills books.** The Deutz Company, Figure 20, retained the original Brons cup, but substituted pump injection of fuel instead of its introduction by the vacuum created in the engine cylinder. They build a 4 and 6 cylinder engine of 4.53-inch bore 6.69-inch stroke, further particulars of which are given in line 11 of Table I. The Benz Company's pre-combustion chamber, Figure 21, has a considerably different design, and it is placed in the center of the cylinder between the inlet and exhaust valves. The fuel pump used on this type of engine is made by the Robert Bosch Company, who realizing the probable reduction in the business of building magnetos developed a very interesting type of fuel pump and injection valve which they intend to furnish engine builders in much the same way they have previously been doing with magnetos and ignition gear. Figure 22 shows the sections of this pump. Here also, as in several other

** Die Patente uber Vorkammer maschinen — F. Ernst Bielefeld. Die Kompressorlose Dieselmaschine, by Ludwig Haasfelder, M. Krayn, Berlin W, 1928.

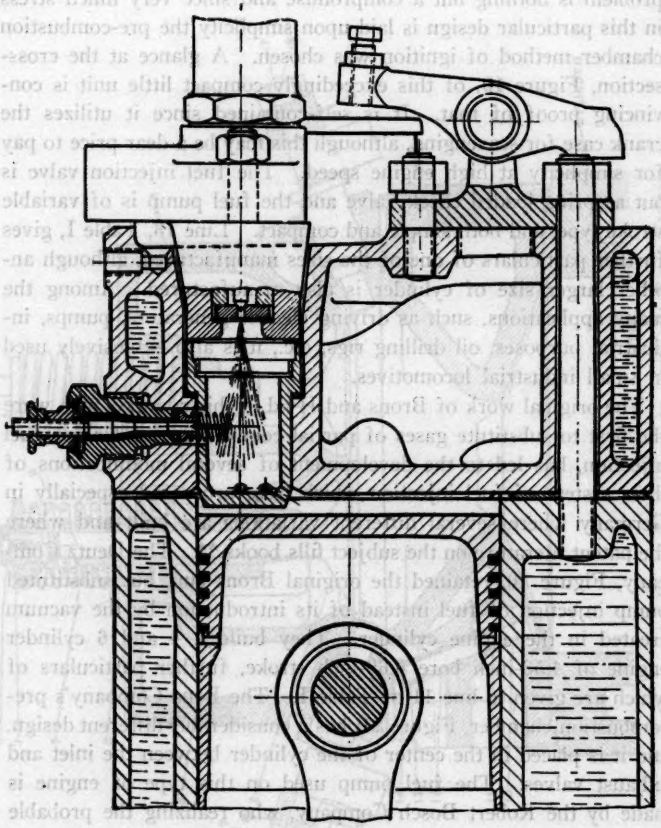


FIGURE 20.—PRECOMBUSTION CHAMBER OF DEUTZ ENGINE.

designs, the suction valve is eliminated and small drilled holes serve as suction ports. They are controlled by the plunger itself. A spring loaded discharge valve of the normal type is fitted and the metering of the charge is regulated by a helical groove in the plunger which has an upper vertical bend as shown on the plunger to the right. The upper movement of the plunger which has a

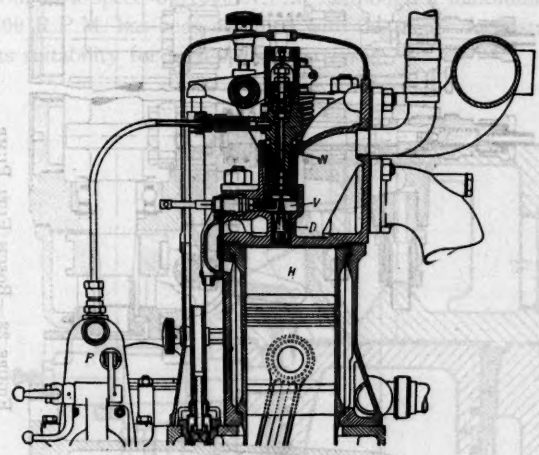


FIGURE 21.—BENZ ENGINE.

constant stroke, forces fuel through the discharge valve up to the moment this helical groove registers with the suction port, when sudden drop of pressure occurs and the further movement of the piston forces the oil above it back into the suction chamber. Rotation of the pump barrel by means of a rack and pinion is done by hand from the operating stand and determines the point on the helical groove which will register with the suction port and therefore the effective stroke. A truck equipped with a six-cylinder M.A.N. engine equipped with Bosch injection made a trial trip from New York to the Road Builders' Convention in Cleveland in January, 1928.[14]

[14] Oil Engine Power, April, 1928, p. 235.

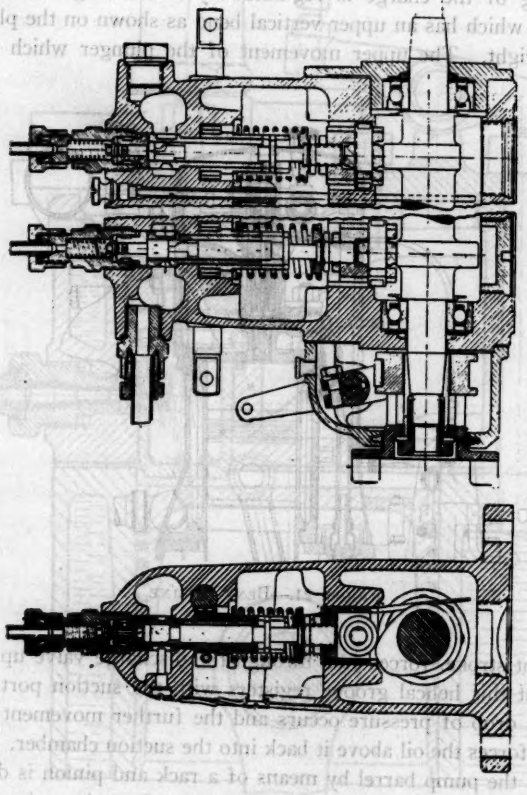


FIGURE 22.—BOSCH FUEL PUMP.

The Bosch Company even goes so far as to develop its own type of engine and Figure 23 shows the latest design of their "Acro" engine of precombustion chamber type. This type of injection depends upon the high velocity of the air returning from the "outside storage" chamber to atomize and distribute the fuel charge it meets. A small unit of 20 B.H.P. of this type is manufactured and has a fuel consumption of .493 pound per B.H.P. per hour at a speed of 1200 R.P.M., although a maximum speed of 2500 R.P.M. has been obtained with this type of construction, and its suitability for high speed is stressed.

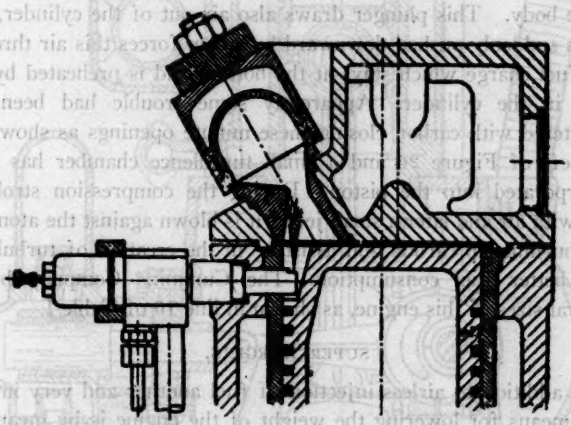


FIGURE 23.—ACRO "OUTSIDE-STORAGE" CHAMBER.

Another German engine using a unique type of precombustion chamber is the so-called Colo-Diesel.[15] A steel plug mounted in the center of the piston enters into the neck of the chamber and through displacer action produces violent turbulence inside the chamber on the compression stroke of the piston, while on the power stroke the heavy pressures of combustion are confined to the chamber alone and a flat top card results. The engine is manufactured in one size of cylinder in numbers from one to six and the details of one are given in line 9 on Table I.

[15] Oil Engine Power, August, 1926, p. 486.

The American development of the same type of engine resulted in the new design of Hill engine, illustrated on Figure 24, which shows the original cup transformed into a cavity through which the fuel spray passes under pressure from the pump. This company manufactures a 5-inch × 7-inch cylinder rated at 12½ B.H.P. at 1000 R.P.M. and a 6-inch × 10-inch cylinder of 20 B.H.P. per cylinder in two, four and six cylinders. See line 14 for further particulars.

The other outgrowth of the Hvid cup is the Cummins engine. Here a low pressure pump is used to charge the fuel valve cavity, shown on Figure 25 in conjunction with the plunger (*c*) in the valve body. This plunger draws also air out of the cylinder, and when suddenly pushed downward by a cam forces this air through the fuel charge which stays at the bottom and is preheated by the heat in the cylinder. Apparently some trouble had been encountered with carbon closing these minute openings as shown on the left of Figure 26 and a small turbulence chamber has been incorporated into the piston. During the compression stroke it fills with air and later a small jet of it is blown against the atomizer tip, burning up carbon and assisting in the creation of turbulence and better fuel consumption. The Cummins Company builds several sizes of this engine, as shown in line 10 of Table I.

SUPERCHARGING.

In addition to airless injection of fuel another and very important means for lowering the weight of the engine is by means of supercharging, i. e., by increasing the supply of available air and oxygen in the cylinder, so that more fuel can be burnt. A blower driven by gears from the crankshaft as shown on Figures 27 and 28 has been used by the Beardmore Company to increase the output of their 12-cylinder V-type engine with 12-inch by 12-inch cylinders from 1500 B.H.P. at 900 R.P.M. to 2000 B.H.P. at 900 R.P.M. The pressure of the supercharging air was 3.25 pounds per square inch. In this case as in all four-cycle engines the supercharging process is also used to scavenge the cylinder of the inevitable residue of combustion gases filling the compression space with very beneficial effect upon exhaust temperature and fuel consumption as shown by Buchi on slow speed engines.

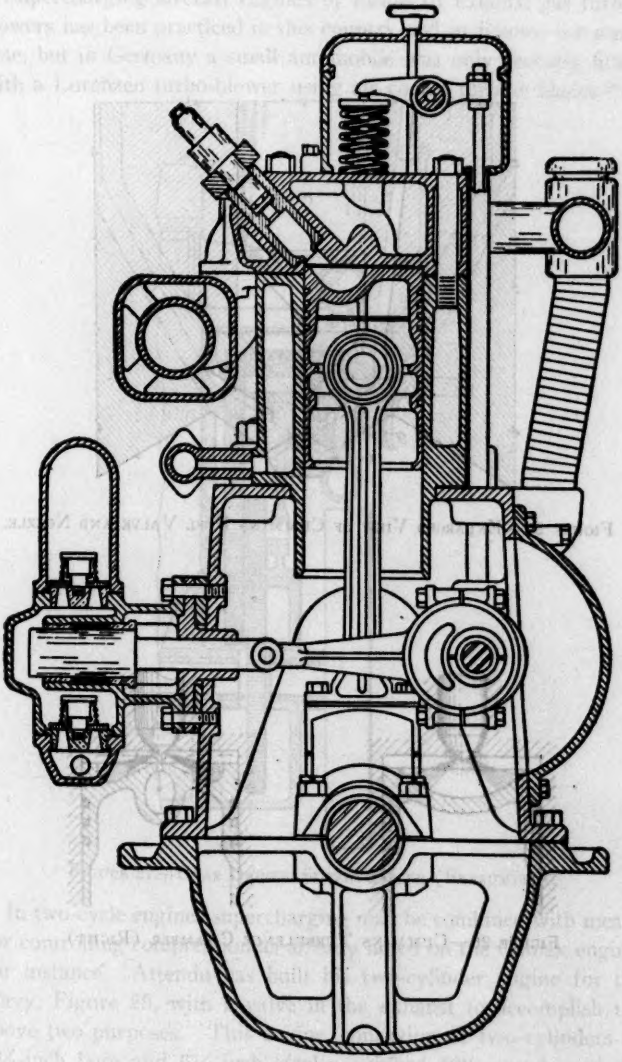


FIGURE 24.—HILL ENGINE.

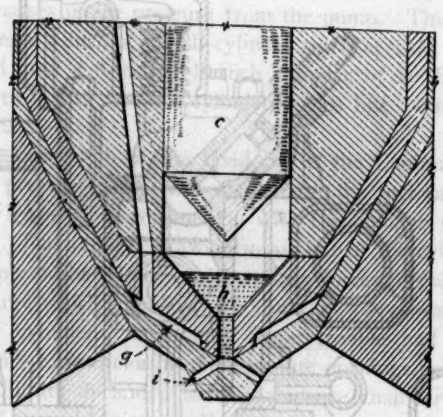


FIGURE 25.—ENLARGED VIEW OF CUMMINS FUEL VALVE AND NOZZLE.

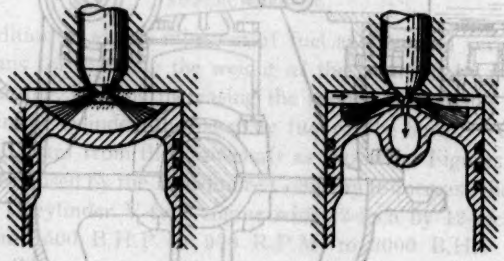


FIGURE 26.—CUMMINS TURBULENCE CHAMBER (RIGHT)

Supercharging aircraft engines by means of exhaust gas turbo-blowers has been practiced in this country and in France for some time, but in Germany a small automobile was only recently fitted with a Lorenzen turbo-blower using air cooled turbine blades.[16]

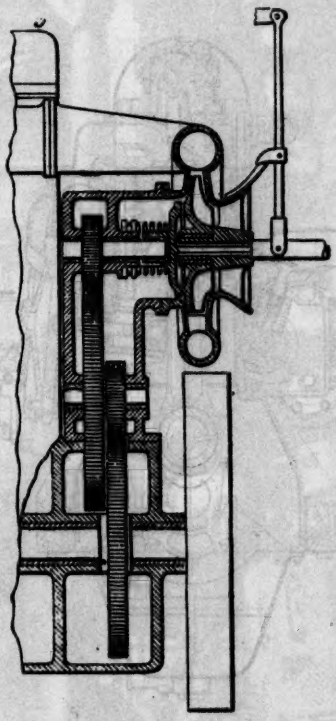


FIGURE 27.—GEAR DRIVEN SUPERCHARGER (BEARDMORE).

In two-cycle engines supercharging may be combined with means for controlling compression as already noted on the Climax engine, for instance. Attendu has built his two-cylinder engine for the Navy, Figure 29, with a valve in the exhaust to accomplish the above two purposes. This engine, consisting of two cylinders of 5½-inch bore and 6½-inch stroke weighed 500 pounds and de-

[16] Mechanical Engineering, March, 1929, p. 220.

veloped 85 B.H.P. at 1620 R.P.M., thus weighing 4.9 pounds per B.H.P. Unfortunately its fuel consumption could not be made low enough and the engine lost one of its important advantages for airship propulsion, namely, cruising radius, and its further development was dropped.

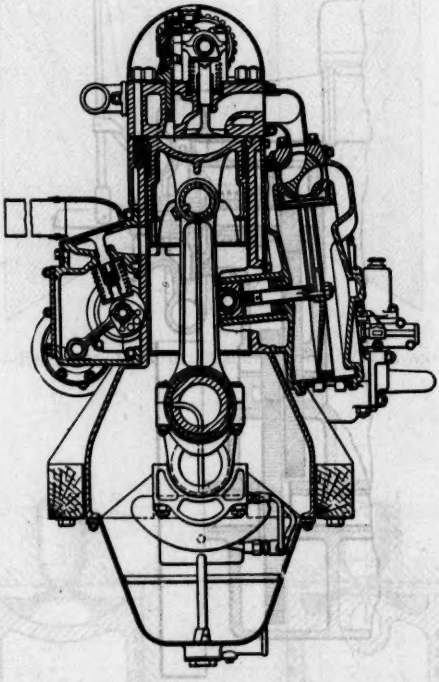


FIGURE 29.—ATTENDU ENGINE (NAVY AIRSHIP).

Supercharging with the Junkers engine is extremely easy of arranging — it means only the affecting of the timing of the upper piston so that the scavenging ports open later than the exhaust and consequently also stay open after the exhaust ports are closed. Any degree of supercharging can be obtained in this manner.

Another two cycle high speed engine that claims as the chief advantage of its design its ability to supercharge, as shown on

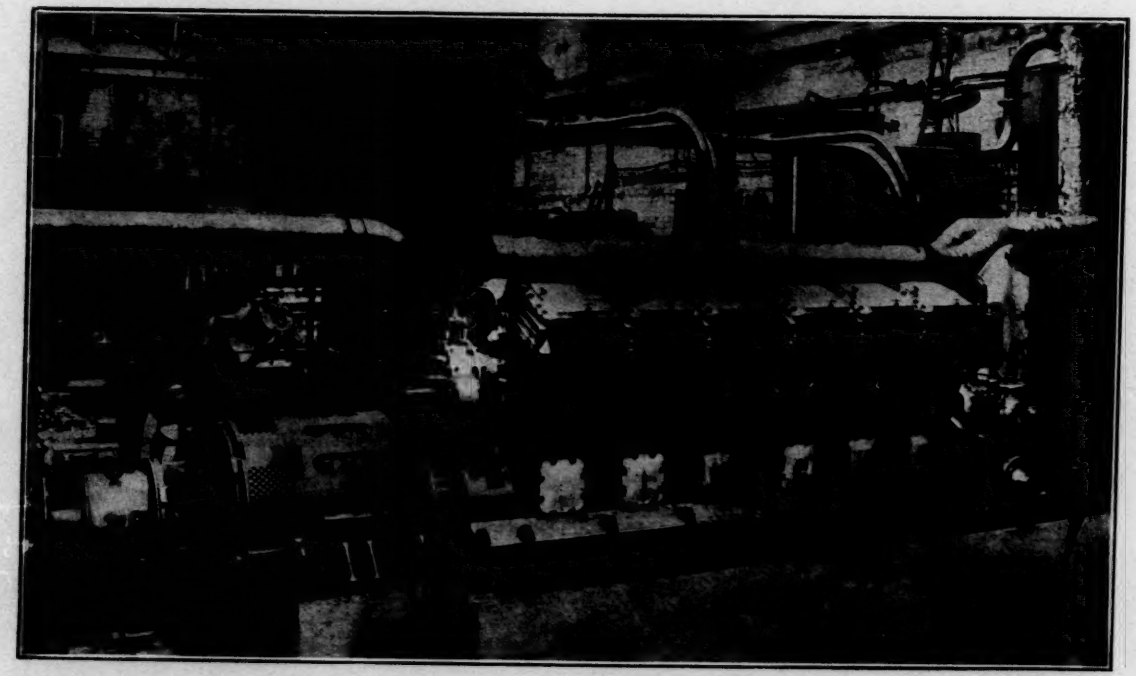


FIGURE 28.—SUPERCHARGED BEARDMORE ENGINE (12 × 12).

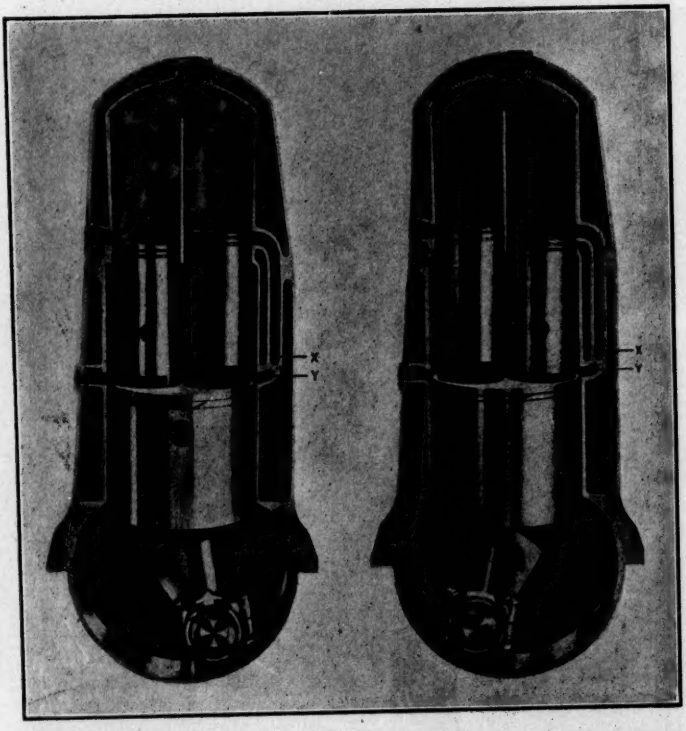


FIGURE 30.—BONANDER ENGINE—EXHAUSTING (LEFT), SUPERCHARGING (RIGHT).

the right side of Figure 30 is the Bonander engine recently developed at the plant of the American Machine and Foundry Company at Brooklyn, N. Y. This engine consists of three units of twin 2¾-inch × 7-inch working cylinders mounted above a common scavenging piston, which is made to serve a similar unit of the engine operating on another crank. A fuel consumption of .50 to .55 pound per B.H.P. per hour is claimed for this engine and a weight of 21.2 pounds per B.H.P. A supercharge pressure of 12 pounds per square inch was used on tests [17] recently made.

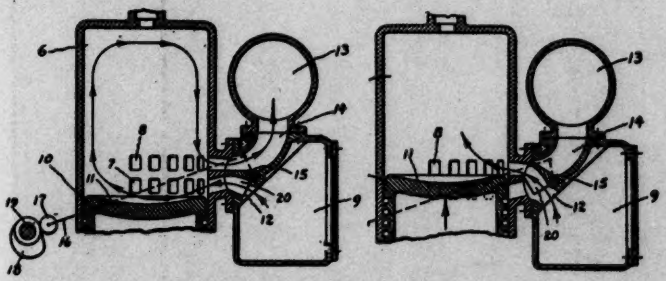


FIGURE 31.—AUTHOR'S PATENTED CONTROL VALVE FOR SUPERCHARGING TWO CYCLE ENGINES WITH TWO ROWS OF SUPERIMPOSED PORTS.

The writer's own recently patented method of supercharging and control of compression applies to two cycle engines using a so-called loop system of scanvenging, which may be like that used by the M. A. N. Company on their large engine or the one Tardieu has utilized for the Peugeout engine [18], line 18, Table I. Figure 31 shows the control valve built into the wall between the scavenging and exhaust ports. On the left it permits normal scavenging to go on while upon the close of the scavenging ports by the piston the exhaust ports are used for supercharging. The control valve can also close the exit of the gases to the exhaust header at any time required by load and speed.

[17] Automotive Industries, February 16, 1929, p. 244.
[18] Oil Engine Power, July, 1926, p. 403.

And finally there is the mysterious "Ersatz Preussen" with its 50,000 horsepower high speed oil engine propelling plant, weighing 17.2 pounds per shaft horsepower for all engineering weight which may or may not add much to the progress of this type of engine.

The writer wishes to take this opportunity to express his profound thanks to the oil engine manufacturers and their Washington representatives who have been kind enough to place at his disposal much of the material embodied in this paper.

TAB

[1]Types of Fuel Injection:—1. Air Injection; 2. Constant Pressure Fluid Injectio

Line	Type of Engine	Cycle	No. of Cylinders	B. H. P. Total	B. H. P. One Cylinder	Bore Ins.	Stroke Ins.	R.P.M.	Displacem't Cubic Ins.	Piston Speed Ft/Min
1	Atlas-Imperial	4	4	80	20	7	8½	650	1308	921
	Atlas-Imperial	4	6	120	20	7	8½	650	1962	921
2	Attendu	2	2	85	42.5	5½	6½	1620	308.9	1755
3	Beardmore	4	8	650	81	8½	12	1000	5140	2000
4	Benz	4	6	70	11.7	4.13	6.5	1300	322.5	1409
5	Bessemer	4	6	200	33.3	7½	10½	650	2782	1138
6	Brotherhood-Ricardo	4	6	300	50	7½	12	1000	3180	1800
7	Buda-MAN	4	4	92	23	6	8	1000	904.5	1333
	Buda-MAN	4	6	180	30	6½	8½	1000	1742	1458
8	Climax-McCarty	2	6	300	50	7½	10	600	2650	1000
9	Colo-Diesel	4	6	60	10	5	7	750	825	875
10	Cummins	4	6	60	10	4½	6	1000	572.5	1000
	Cummins	4	6	170	28.3	6½	9	800	1931	1200
	Cummins	4	6	500	83.3	10	12	800	5655	1600
11	Deutz	4	6	85	14.1	4.53	6.69	1250	646	1394
12	Dorner	4	4	30	7.5	3.74	6.30	1000	277	1048
13	Fairbanks-Morse	2	4	60	15	6	6½	800	735	867
14	Hill	4	6	75	12.5	5	7	1000	824.7	1166
	Hill	4	6	120	20	6	10	800	1696.4	1333
15	Ingersoll-Rand	4	6	300	50	10	12	550	5653	1100
	Ingersoll-Rand	4	6	825	137.5	14½	16	500	16400	1333
16	Junkers	2	2 opp.	65	32 5	3.15	7.9+7.9	1000	246.1	1316
17	Maybach	4	6	150	25	5½	7½	1300	1015	1544
18	Pengeot	2	2	53	26.5	4.7	5.9	1450	204.8	1426
19	"Speedway"	4	6	300	50	8½	11	700	3747	1284
20	Standard M. C.	4	4	60	15	6	8	650	904.5	867
	Standard M. C.	4	6	130	21.6	6	8	900	1356	1200
21	Treiber	4	12v	3000	250	16	16	700	38600	1867
	Treiber	4	12v	750	62.5	9½	12	700	10210	1400
22	Westinghouse	4	6	300	50	8½	12	800	3850	1600
	Westinghouse	4	12v	1340	111.6	12	12	800	16290	1600
23	Winton	4	6	150	25	7	9½	750	1963	1062
	Winton	4	6	250	41.6	8	10	750	3019	1250
	Winton	4	8	425	53	8½	12	700	5445	1400
24	Koerting	4	6	90	15.0	5.1	7.1	1200	880	1420
25	Linke-Hoffmann	4	4	50	12.5	4.5	6.3	1200	410	1200
26										
27										
28										

TABLE I.

...id Injection; 3. Direct Pump Fluid Injection; 4. Pre-Combustion Chamber Fuel Injection.

	Piston Speed Ft/Min	Brake M.E.P. Lbs. per Sq. In.	Specific Volume Cu. Ft. per B.H.P. Min.	Weight Lbs. Total	Weight Lbs. Per B.H.P.	Weight Lbs. per Cu. In. of Pist. Displ.	Fuel Consumption per B.H.P. Hour	Type of Fuel Injection	Remarks
...8	921	74.5	6.15	4600	57.5	3.52	.42	2	
...2	921	74.5	6.15	6600	55.0	3.36	.42	2	
...9	1755	67.4	3.41	477	4.9	1.54	....	3	
...0	2000	107.5	4.58	4600*	7.1	.895	.385	3	*3600 -Alum. Crankcase
...5	1409	81.6	5.62	1600	22.9	3.06	.456 / .558	4	
...2	1138	87.6	5.24	8200**	41	2.95	.43 / .45	2	**10,000 in Cast Iron
...0	1800	74.7	6.14	12000	40	3.78	.38	3	
...5	1333	80.6	5.69	3200	34.8	3.54	.435	3	
...2	1458	81.85	5 6	5425	30.3	3.12	.435	3	
...0	1000	74.8	3.07	12000	40	4.53	....	1*	*Hot Air Fuel Injection
...5	875	76.8	5.97	2200	36.6	2.67	....	4	
...5	1000	83.0	5.52	2200	36.7	3.84	....	4	
...1	1200	87.2	5.26	5000	29.4	2.59	....	4	
...5	1600	87.6	5.24	14500	29.0	2.56	....	4	
...6	1394	83.3	5.5	1632	19.2	2.53	.425	4	
...77	1048	85.8	5.34	....	....	....	.479	3	
...35	867	40.4	5.68	3940	65.7	5.36	....	4	
...7	1166	72.0	6.37	3920†	50.0	4.75	.5	4	†Includes Reverse
...4	1333	70.0	6.54	7500†	58 6	4.42	.5	4	Gear and Electric Start
...53	1100	76.4	6.0	19000	63.3	3.36	.44	3A	Opposed Sprays
...00	1333	79.7	5.76	45000	54.6	2.74	.44	3A	Price System
...5.1	1316	104.5	2.19	610	9.5	2.48	....	3	
...15	1544	90.1	5.09	2650	17.7	2.61	.40 / .49	1	
...4.8	1426	70.5	3.25	530	10	2.59	....	3	
...747	1284	90.2	5.06	7600	25.3	2.03	....	2	
...4.5	867	80.9	5.67	3500	58.3	3.87	.43	2	
...156	1200	84.4	5.44	5110	39.3	3.78	.41	2	
...500	1867	87.8	5.22	63200	21.1	1.64	.42	2	
...210	1400	83.1	5.52	18500	24.7	1.81	.42	2	
...350	1600	76.0	5.94	12000	40.0	3.12	.40	3	
...290	1600	81.4	5.63	33300	25 0	2.06	.40	3	
...963	1062	80.7	5.68	5700	38	2.9	.44	2	
...619	1250	87.5	5.24	7400	29.6	2.45	.44	2	
...445	1400	88.3	5.19	13000	30.5	2.39	.44	2	
...880	1420	67.5	6.79	3360	36.0	3.75	....	4	
...410	1300	80	5.7	1100	22.0	2.7	....	3	

Line	Type of Engine	Cycle	No. of Cylinders	R. H. P.		Bore	Stroke	R.P.M.	Displacement Cubic Ins.	Piston Speed
				Total	One Cylinder	Ins.	Ins.			Ft/Min
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	Atlas-Imperial	4	6	120	20	7	8¼	650	1962	921
2	Attendu	2	2	85	42.5	5½	6½	1620	308.9	1755
3	Beardmore	4	8	650	81	8½	12	1000	5140	2000
4	Benz	4	6	70	11.7	4.13	6.5	1300	522.5	1409
5	Bessemer	4	6	200	33.3	7½	10½	650	2782	1138
6	Brotherhood-Ricardo	4	6	300	50	7½	12	1000	3180	1800
7	Buda-MAN	4	4	92	23	6	8	1000	904.3	1333
	Buda-MAN	4	6	180	30	6½	8½	1000	1743	1458
8	Climax-McCarty	2	6	300	50	7½	10	600	2650	1000
9	Colo-Diesel	4	6	60	10	5	7	750	825	875
10	Cummins	4	6	60	10	4½	6	1000	572.5	1000
	Cummins	4	6	170	28.3	6½	9	800	1931	1200
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11	Deutz	4	6	85	14.1	4.53	6.69	1250	646	1394
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14	Hill	4	6	75	12.5	5	7	1000	824.7	1166
	Hill	4	6	120	20	6	10	800	1696.4	1333
15	Ingersoll-Rand	4	6	300	50	10	12	550	5653	1100
	Ingersoll-Rand	4	6	825	137.5	14½	16	500	16400	1333
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17	Maybach	4	6	150	25	5½	7½	1300	1015	1544
18	Peugeot	2	2	53	26.5	4.7	5.9	1450	204.8	1426
19	"Speedway"	4	6	300	50	8½	11	700	3747	1284
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	Standard M. C.	4	6	130	21.6	6	8	900	1356	1200
21	Treiber	4	12v	3000	250	16	16	700	38600	1867
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	Winton	4	6	250	41.6	8	10	750	3019	1250
	Winton	4	8	425	53	8½	12	700	5445	1400
24	Koerting	4	6	90	15.0	5.1	7.1	1900	880	1420
25	Linke-Hoffmann	4	4	50	12.5	4.5	6.5	1200	410	1300
26										
27										
28										

	Piston Speed Ft/Min	B.M.E.P. M.E.P. Lbs. per Sq. In.	Volume Cu. Ft. per B.H.P. Min.	Lbs. Total	Lbs. Per B.H.P.	Lbs. per Cu. In. of Pist. Displ.	Consumption per B.H.P. Hour	Type of Fuel Injection	Remarks
308	921	74.5	6.15	4600	57.5	3.52	.42	2	
962	921	74.5	6.15	6600	55.0	3.36	.42	2	
8.9	1735	67.4	3.41	477	4.9	1.54	....	3	
140	2000	107.5	4.58	4600*	7.1	.895	.385	3	*3600 -Alum. Crankcase
2.5	1409	81.6	5.62	1600	22.9	3.06	{.456 .558}	4	
782	1138	87.6	5.24	8200**	41	2.95	{.43 .45}	2	**10,000 in Cast Iron
180	1800	74.7	6.14	12000	40	3.78	.38	3	
4.5	1333	80.6	5.69	3200	34.8	3.54	.435	3	
743	1458	81.85	5 6	5435	30.3	3.12	.435	3	
650	1000	74.8	3.07	12000	40	4.53	....	1*	*Hot Air Fuel Injection
825	875	76.8	5.97	2200	36.6	2.67	....	4	
72.5	1000	83.0	5.52	2200	36.7	3.84	....	4	
931	1200	87.2	5.26	5000	29.4	2.59	....	4	
655	1600	87.6	5.24	14500	29.0	2.56	....	4	
646	1394	83.3	5.5	1632	19.2	2.53	.425	4	
277	1048	85.8	5.34	....	....	....	.479	3	
735	867	40.4	5.68	3940	65.7	5.36	....	4	
24.7	1166	72.0	6.37	3920†	50.0	4.75	.5	4	†Includes Reverse
96.4	1333	70.0	6.54	7500†	58 6	4.42	.5	4	Gear and Electric Start
3653	1100	76.4	6.0	19000	63.3	3.36	.44	3A	Opposed Sprays
5400	1333	79.7	5.76	45000	54.6	2.74	.44	3A	Price System
46.1	1316	104.5	2.19	610	9.5	2.48	....	3	
1015	1544	90.1	5.09	2650	17.7	2.61	{.40 .49}	1	
04.8	1426	70.5	3.25	530	10	2.59	....	3	
3747	1284	90.2	5.06	7600	25.3	2.03	....	2	
04.5	867	80.9	5.67	3500	56.3	3.87	.43	2	
1356	1200	84.4	5.44	5110	39.3	3.78	.41	2	
8600	1867	87.8	5.22	63200	21.1	1.64	.42	2	
0210	1400	83.1	5.52	18500	24.7	1.81	.42	2	
3850	1600	76.0	5.94	12000	40.0	3.12	.40	3	
6290	1600	81.4	5.63	33500	25 0	2.06	.40	3	
1963	1062	80.7	5.68	5700	38	2.9	.44	2	
3019	1350	87.5	5.34	7400	29.6	2.45	.44	2	
5445	1400	88.3	5.19	13000	30.5	2.39	.44	2	
880	1420	67.5	6.79	3300	36.0	3.75	....	4	
410	1300	80	5.7	1100	22.0	2.7	....	3	

INSTALLATION OF A WATER RECOVERY SYSTEM AT THE FUEL OIL TESTING PLANT.

By Commander J. J. Broshek,* U. S. Navy, Member.

The Fuel Oil Testing Plant is maintained by the Bureau of Engineering of the Navy Department for the purpose of testing boilers, equipment, apparatus and materials used in boiler operation and for working out problems connected with the management of boilers and fuel oil burning. In order to carry on this work a number of boilers are installed similar to those found on board vessels of the Navy together with the auxiliaries necessary for their operation and instruments needed for the obtaining of the required data. As most tests are conducted to determine the effect on evaporative efficiency either of new equipment or changed methods of operating, the boilers are under steam a great deal at various rates of firing, and all of the steam generated is discharged directly to the atmosphere. Philadelphia city water which is used for feed is notoriously bad for boilers because it contains a large percentage of scale forming impurities which when deposited on the heating surfaces cause overheating of the metal, loss in efficiency and are extremely difficult to remove. Furthermore the practice of using one hundred per cent (100 per cent) of raw make up feed water does not simulate service conditions where the steam is condensed and only a small proportion of make up feed is used and even that is free from impurities having been made in the ship's evaporating plant. Because of being obliged to use raw Philadelphia water as feed the Fuel Oil Testing Plant has, since its establishment, been handicapped in its work as the boilers require frequent cleaning, tubes burn out especially at high rates of evaporation and even when the water is treated scale deposition cannot be prevented and foaming and priming occur after a short period of operation.

* Officer-in-Charge, Fuel Oil Testing Plant, Navy Yard, Philadelphia, Pa.

The need of a condensing or water recovery system has long been recognized, several plans have been drawn up and the whole proposition given very careful study and thought. The problem of testing a boiler at 600 pounds pressure and 200 degrees superheat, difficulties experienced in running tests of the Plant's 12,000 square foot boiler at high rates of evaporation and the desirability of more closely simulating service conditions were the principal factors which forced the issue and led to the undertaking of the project at the present time.

The purpose of a water recovery system for the Fuel Oil Testing Plant is, as its name implies, the recovery of the steam generated in the boilers so that it may be used over again as feed. In working up the details of the installation the following considerations were kept in mind:

(*a*) The maximum quantity of steam to be handled is 225,000 pounds per hour at 300 pounds pressure 150 degrees superheat but provision must be made for 100,000 pounds at 600 pounds pressure and 200 degrees superheat.

(*b*) Funds for carrying the project into execution are limited—surplus and used material must be used as far as possible.

(*c*) The installation must be safe.

(*d*) Condensing equipment need not operate under a vacuum—the steam need be condensed only.

(*e*) The auxiliaries use about 25,000 pounds of steam per hour.

(*f*) The circulating water must be pumped from the reserve basin and discharged to the back channel, both a considerable distance from the pumps, and the water must be lifted about 15 feet.

After making a survey of materials and equipment available for the project at various Navy Yards, a number of feed water heaters manufactured by the Griscom Russell Company were located which it was decided could be used for condensers. They are four pass straight tube heaters and were designed to raise the temperature of 587,000 pounds of water from 90 degrees Fahrenheit to 225 degrees Fahrenheit when supplied with sufficient steam at 10 pounds pressure. For use as condensers they were altered by being made two pass instead of four pass, the cooling water inlet and outlet nozzles were made equal in area to the combined area of the

tube bank and the inlet nozzle was also changed to provide a free flow without sharp change in direction. Curves showing the condensing capacity, pressure drop and circulating water required as calculated by engineers of the Griscom Russell Company are shown in Figure I. It will be noted that with 3000 gallons of circulating water per minute over 100,000 pounds of steam will be condensed per hour with a pressure drop through the tubes of slightly less than 10 pounds. Two condensers are therefore being provided, the combined capacity of which is over 200,000 pounds of steam per hour, which though somewhat less than the maximum boiler output is considered adequate since provision is also being made for bleeding steam to the atmosphere in case of necessity at very high rates and the erection of a third condenser would greatly complicate the installation as well as materially increase the cost.

It is expected that the steam will be condensed at a temperature corresponding to a pressure of 10 pounds or 240 degrees F. Obviously this temperature must be lowered before the condensate can be discharged into the hot well or storage tank from which it is to be pumped to the weighing tanks and the boilers. If the water were hot enough to vaporize when pumped into the weighing tanks an error in the weight of water fed to the boiler would be introduced by the quantity vaporized. It is considered that the temperature of the water should be not more than 150 degrees F. and provision for cooling the condensate to this temperature is made by the installation of a cooler which is simply a third feed water heater through which the condensate from the condensers is passed. The cooler, like the condensers, is made two pass but unlike them it is drained through the normal steam inlet which forms an excellent reservoir or well to which the condensate pump suction is connected. Approximately 225,000 pounds of condensate will have to be cooled per hour at maximum capacity from 240 degrees F. to 150 degrees F. and provision has been made for supplying 1000 gallons of circulating water per minute or over two pounds of circulating water per pound of condensate.

The circulating water is drawn from the reserve basin a distance of 420 feet from the pump suctions and is discharged through a 600-foot line into the back channel. It was at first contemplated running both suction and discharge to the reserve basin on account

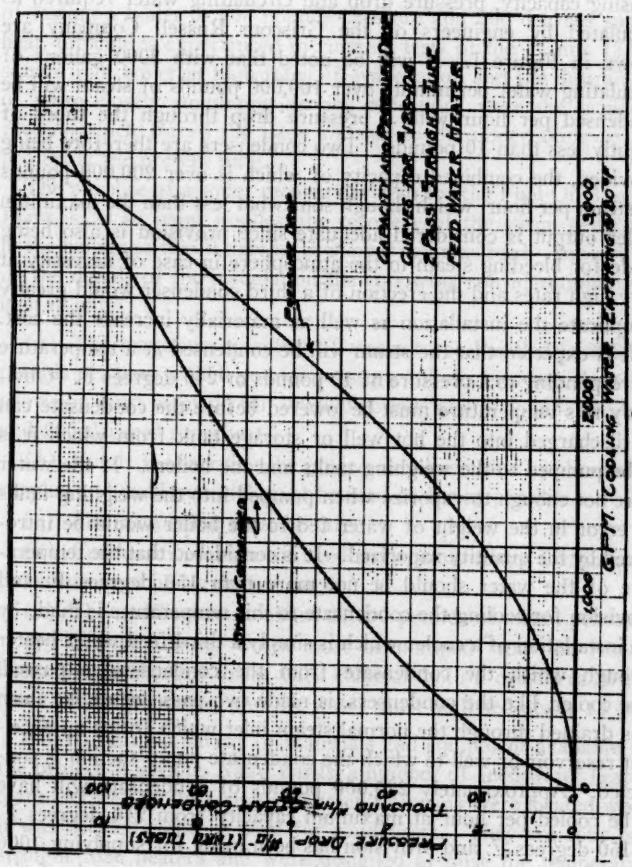
Curves for "Zero-Pressure Drop" 2-Pass Straight Tube Feed Water Heater
Pressure Drop
Pressure Gradient
1000 2000 3000 4000 G.P.M. Cooling Water Entering at 70°
Pressure Drop lbs/in² (Tube Tubes)
Thousand lbs/hr Steam Condensed
0 10 20 30 40

Fig. 1

of the shorter distance but there is very little movement of the water here and it was calculated that with the quantity of water used the temperature in the basin would rise to a prohibitive degree after less than a day's running. On account of the long leads of suction and discharge piping the friction loss is kept at a minimum by using pipe of ample diameter and hence the velocity through the piping is low. It was decided to use 20-inch bell and spigot cast iron pipe for the suction and discharge lines because it is comparatively easy to install, is very durable and primarily because it is the cheapest. Terra cotta pipe was considered for the discharge line but because of the comparatively high temperature of the discharge water the idea of using anything but iron pipe was given up.

The most suitable pumps available were 14-inch turbine driven condenser circulating pumps manufactured by the Worthington Pump and Machinery Corporation, driven by Terry turbines originally intended for installation on Eagle boats. These pumps have a rated capacity of 3500 G.P.M. at 1200 R.P.M., the Capacity-Head curves of which are shown in Figure II. It was decided to install two units to provide the necessary pumping capacity, 6000 G.P.M. for the two condensers and 1000 G.P.M. for the cooler, and to split the suction line providing for a separate 14-inch suction to each pump. Water to the cooler is supplied by branches from the discharge of each pump thus eliminating the need of installing a separate pump for the cooler. The entire circulating system is fitted with adequate valves and fittings so that any unit may be shut down when operating at reduced rates. The pump suctions are 13½ feet above mean low water and about 8 feet above mean high water, the friction head loss in the suction piping and fittings is 3 feet, so under the worst conditions a suction lift of 16½ feet has to be provided for. As this is a severe condition for a circulating pump designed for use on a ship, the installation is being equipped with priming or air exhausting apparatus which consists simply of mounting air ejectors on the highest point of the system.

It has been calculated that the total head under which the pumps will operate at maximum capacity will be 13½ feet suction lift, 3 feet friction loss in the suction line and fittings, 20 feet friction loss in the condensers and 9 feet of discharge head through the pump to the top of the condensers and discharge line, making a

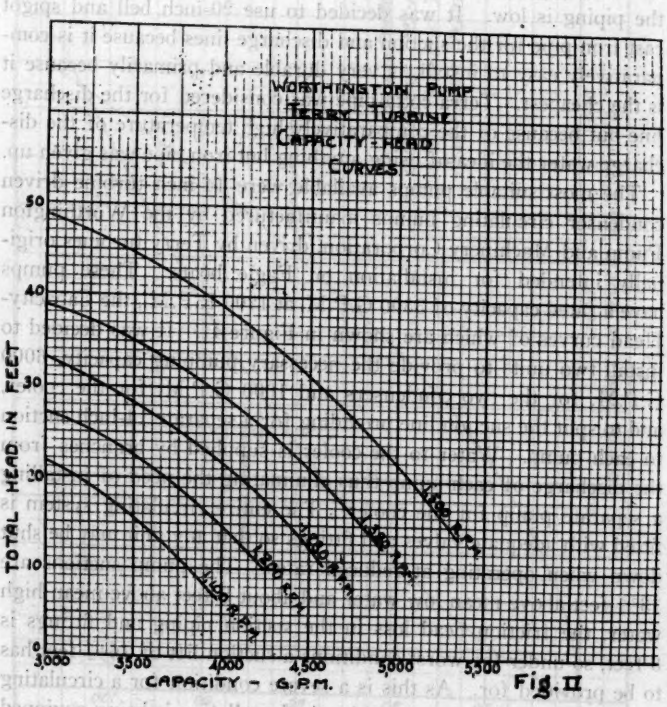
WORTHINGTON PUMP
TERRY TURBINE
CAPACITY - HEAD
CURVES
TOTAL HEAD IN FEET
50
40
30
20
10
0
3,000
3,500
4,000
4,500
5,000
5,500
CAPACITY - G.P.M.
1600 R.P.M.
1500 R.P.M.
1400 R.P.M.
1300 R.P.M.
1200 R.P.M.

Fig. II

total head of about 45½ feet. The greatest problem is the suction lift which is considerable for this type of pump, but it is expected that the air ejecting apparatus will assist materially in overcoming the difficulty and in maintaining the flow after suction has been established.

As previously stated, one of the governing considerations in the design of the installation was that capacity should be provided for 225,000 pounds of steam per hour at 300 pounds pressure and 150 degrees superheat. This represents the output of the Plant's 12,000 square foot express type boiler which is already installed with all necessary auxiliaries, piping and bleeder connection. The line to the condensers is being led from the 8-inch stream line already provided. A high pressure boiler having 5000 square feet of heating surface capable of generating 100,000 pounds of steam at 600 pounds pressure and 157 degrees superheat is also being connected to the system. As the quantity of steam produced by the high pressure boiler is so much less than the output of the 12,000 square foot boiler it is only necessary to provide for the higher pressure and higher temperature. At "A" in Figure III is a manually operated control valve by means of which an operator regulates the steam output by keeping the boiler pressure constant, opening or closing the valve as the amount of steam generated varies either with change of rate or because of minor fluctuations. For the high pressure boiler two control valves will be used, one at "B" to keep 600 pounds on the boiler and the other at "A" to maintain 300 pounds on the auxiliary steam line, the steam being expanded in the line from 600 pounds at valve "B" to 300 pounds at valve "A." A safety valve "C" is installed on the low pressure side of control valve "B" to protect the 300-pound main and auxiliary steam line in case the pressure should get out of control for any reason. Only one boiler will be in operation at a time.

After leaving the control valve "A" the pressure of the steam is reduced by expansion through a series of pipes which gradually increase in diameter to accommodate the increased volume. In the design of the steam line to the condensers certain limitations presented themselves which had to be given due consideration, chief among which were that the steam inlet nozzles to the condensers were 22 inches in diameter or equal in combined area to a 30-inch

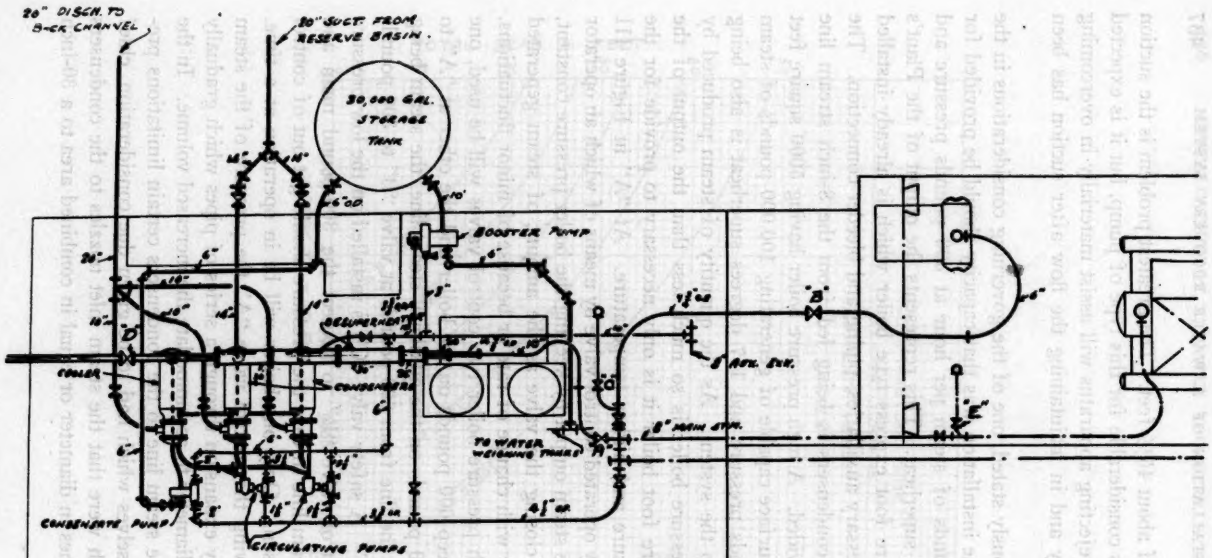

Fig. III

WATER RECOVERY SYSTEM
FUEL OIL TESTING PLANT

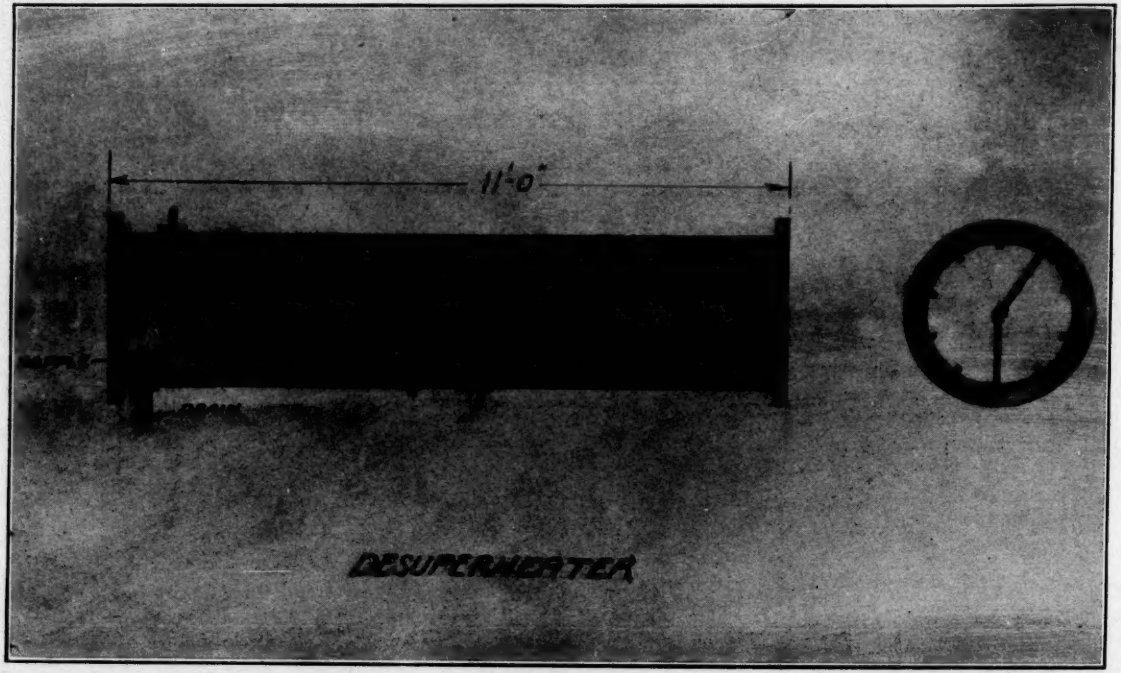

FIGURE IV.

pipe, that the line from the 12,000 square foot boiler was already installed and that operation at maximum capacity would be infrequent. It would have been highly desirable for the steam line from the low pressure boiler to have been larger than 8 inches on account of the high steam velocities that will result at maximum rates, but due to limited funds it is necessary to use the existing installation as far as possible. This boiler is not yet equipped with a superheater, which will be installed at some later date when superheater tests are undertaken. The original installation also provides for the bleeding of steam from this boiler to the atmosphere at the control valve "E."

It is estimated that 25,000 pounds of steam will be used per hour by all the auxiliaries so that under the most severe condition 200,000 pounds of steam per hour at 300 pounds pressure and 570 degrees total temperature will go to the condensers. Based on this condition the pressures to be expected in the various stages of the expansion piping have been calculated to be as follows:—135 pounds in the 14-inch section, 60 pounds in the 20-inch, 35 pounds in the 24-inch, 15 pounds at the desuperheater inlet and 10 pounds in the 30-inch connection to the condensers.

In expanding the steam pressure down in this manner it is superheated above the original 150 degrees so that at the desuperheater inlet at a pressure of 15 pounds the temperature is 530 degrees, which represents 275 degrees of superheat. The total heat of the steam, 1295 B.T.U. per pound, has not changed. Because the condensers will not stand such high temperatures, the desuperheater is being provided to remove the superheat. To keep down the cost of construction it is made very simple and consists only of an 11-foot length of 30-inch pipe flanged at the ends. Along its length and spaced equally around its circumference are installed spray nozzles. The first and sixth spray nozzles extend to the center of the desuperheater and spray directly into the path of the steam while the others spray at right angles to the path as shown in Figure IV. The spray nozzles are simply modified fuel oil atomizers.

FIGURE IV.

The determination of the amount of spray required is comparatively easy. The total heat of the steam is 1295 B.T.U.s, its original value at 300 pounds pressure and 150 degrees superheat, since

practically none of the heat has been lost by expansion. This steam is cooled to a temperature corresponding to a saturated pressure of 10 pounds, the heat content of which is 1160 B.T.U.s, so that it is necessary to provide enough water to absorb 135 B.T.U.s from each pound of steam. In absorbing this 135 B.T.U.s the spray water is itself evaporated and forms steam at 10 pounds saturated pressure. If the temperature of the spray water is 150 degrees F. and it is evaporated to dry steam at 10 pounds pressure, each pound will absorb the difference between its initial and final heat content or 1042 B.T.U.s Therefore each pound of spray water will desuperheat almost 7¾ pounds of steam or for 200,000 pounds of steam 26,000 pounds of spray water at 150 degrees F. are required. The quantity of steam going to the condensers is increased by the amount of spray water evaporated or is now 226,000 pounds per hour.

The spray water is discharged by a separate pump at about 50 pounds pressure to a manifold—similar to a fuel oil burner manifold—from which connections are made to the individual spray nozzles. There is a separate cut out valve for each so that the number of nozzles in use or the amount of spray water supplied can be very closely regulated. A baffle and ample sized drain is installed to collect and carry off any water which may form and suitable gauges and thermometers are being fitted to assist in regulating the water supply.

At maximum rating the condensers will be forced to the limit of their capacity and even then it is not expected that they will be able to handle all the steam. An inspection of the curves in Figure 1 shows that with 3000 gallons of circulating water per minute 103,500 pounds of steam will be condensed per hour in each unit based on an injection temperature of 80 degrees F. With cooler water the quantity of steam condensed will be somewhat greater. Since the amount of steam is 226,000 pounds, which will be further increased by the amount of auxiliary exhaust not used in the feed water heaters, it will be necessary to bleed off a certain portion when steaming at very high rates of evaporation. Provision for this contingency is made by the installation of a 30,000-gallon storage tank from which the water is taken for feed. If this tank is full at the beginning of a run and 40,000 pounds are bled off

per hour, there will be enough make up supply to run for six hours at maximum capacity without resorting to the use of city water for make up feed. Eventually it is expected that an evaporating plant will be installed to use this excess of steam in making the necessary make up feed for this system. For the present it is contemplated using one of the Plant's older boilers for generating steam from city water to provide make up. If the boiler is run at a low rate and the water properly treated very little harmful scale deposit is expected.

If 207,000 pounds of steam go to the condenser per hour and 6000 gallons of circulating water are supplied per minute, the ratio of water to steam is about 14½ to 1. Since 952 B.T.U.s have to be absorbed from each pound of steam to condense it, each pound of circulating water will take up about 65 B.T.U.s and the temperature will rise from 80 degrees F. to 145 degrees F.

After leaving the condensers the condensate drains to the coolers where its temperature is lowered and from which it is pumped by the condensate pump and discharged to the storage tank. From the storage tank the water is taken to the booster pump and discharged to the water weighing tanks and from there the existing installation of feed pumps, feed heaters and lines take the water to the boilers.

This installation is unique because of the fact that no effort is made to secure economy of operation; rather, every effort is made to use as much steam as possible before it is condensed provided the condensate can be saved. Provision is made for a wide operating range to accommodate the various rates used in running evaporation tests of boilers. To insure safety to personnel, the system is equipped with ample relieving devices in case there should be a sudden rise in pressure due to stoppage of any unit or for any other cause. The safety valve at "C" provides for a rise in pressure from the high pressure boiler and an atmospheric relief valve at "D" in Figure III takes care of any abnormal rise on the low pressure end of the system. In addition, the valve at "D" and another at "E" provide for bleeding to the atmosphere in case of overload or in case the condensing equipment should be out of service. Suitable cut out valves on the steam line to the condensers are installed so that either or both may be isolated from the system.

The operation of the high pressure boiler will involve no difficulties. Its maximum steam output is 100,000 pounds per hour, the heat content of which is only about 25 B.T.U.s per pound more than that of the steam from which the calculations were made. Because of the high pressure at which this boiler operates none of the feed system at present installed can be used, so a special high pressure feed pump and feed heater are being erected. The feed pump turbine will use steam at 600 pounds pressure which will be branched from the high pressure side of the control valve. Extra heavy pipe and fittings suitable for the 600 pounds are being provided.

The other boilers in the Plant are also being joined to the system by means of a line connecting the old and new steam mains through which the steam generated in the old boiler plant will pass to the condensers and return of the condensate is taken care of by installing a line from the booster pump discharge to the water weighing tanks in the old boiler room. When running the old plant, all the steam used by the auxiliaries will be lost since no exhaust line is being installed at present.

To prevent any part of the system — both steam and water — from getting air bound a complete air ejecting installation is being made with connections to all parts where air pockets are likely to form. Gauges and thermometers are being provided wherever necessary to give a clear indication of pressure and temperature conditions throughout the system. All drains from feed water heaters, machinery and pipe lines are being connected to the storage tank to prevent loss of fresh water. The circulating water suction is fitted with a screen of ⅜ mesh and of ample area which can be removed for cleaning in case it becomes clogged with sediment and a fitting is provided at the pumps to which connection can be made to the Yard's fire main so that the whole suction line may be flushed if silt or mud should deposit in it. The discharge is fitted with a baffle or deflector to prevent the possibility of enough earth being scoured out to cause the sea wall in the immediate vicinity to cave in.

With the exception of the 20-inch circulating water piping, the 600-pound lines and fittings, the high pressure feed pump and heater and the 30- and 20-inch sections of the steam line to the condensers, no material is being purchased for the project, all the

rest having been obtained from surplus stocks or salvage. Every effort was made to utilize available material as far as possible and to employ the least costly methods of installation and fabrication. It is not an ideal installation but is the best that could be done under the circumstances of limited funds and the necessity of using available equipment. The project has not yet been completed but the work is progressing steadily; it will be interesting to see just how the actual performance checks with the calculated expectations. Some changes may be required and additions made but in any event the Fuel Oil Testing Plant has a water recovery system which it has urgently needed ever since it was established,

EFFICIENCY EFFECT OF EROSION OF BLADES ON A RADIAL FLOW TURBINE.

By Burnett F. Treat, Associate Member.*

In connection with discussion of material required for turbine blading, there arose the question of the effect on steam consumption of erosion of the blade tips on re-entrant type turbines in which the steam enters in a radial direction as distinguished from those in which steam enters axially. In these so-called radial-flow turbines the blades are ordinarily milled on the periphery of the wheel; therefore, any specification calling for special steel or alloy blades would ordinarily require the entire rotor to be made of such special metal.

The following article describes and gives results of a test made at the U. S. Naval Engineering Experiment Station on a turbine to determine steam consumption with progressive simulated blade erosion.

The turbine tested was a 24-inch diameter wheel, designed for normal operation at 72 brake horsepower, 250 pounds per square inch steam pressure, and 1820 R.P.M. speed. This turbine as set up for test is shown with the wheel exposed on Figure 1. The wheel of this turbine can be thought of as a worm gear, and in operation this gear is driven by three worms of steam. Steam issues from a nozzle (at the lower end of the casting on which the steel scale is resting in the photograph) to strike into the left side of one or several of the steam buckets between blades. In these grooves the steam has its direction of travel reversed to issue in a plane parallel to the side at the right-hand side blade shrouding. The steam is successively redirected in the reversing chamber (the casting mentioned) to make several screw turns until the kinetic energy available at the nozzle has entirely exerted itself as a turning

* Associate Mechanical Engineer, U. S. Naval Experiment Station.

FIGURE 1.—THE TEST TURBINE AS RECEIVED.

FIGURE 2.—A TURBINE ROTOR AFTER TWELVE YEARS' ACTUAL SERVICE.

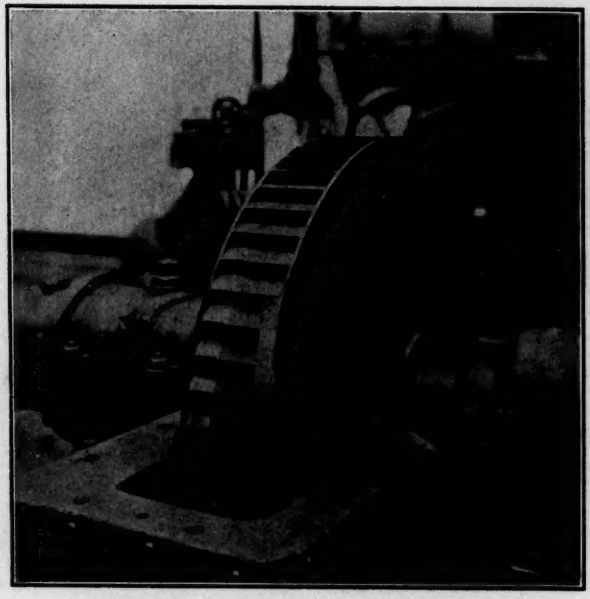

FIGURE 3.—AFTER FIRST CUT TO SIMULATE EROSION.

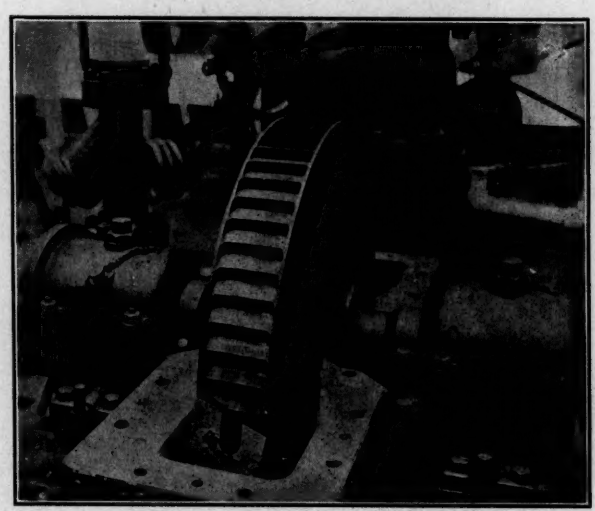

FIGURE 4.—FINAL CONDITION, CUT ACROSS BLADE.

effort on the face of the wheel. This particular turbine has three nozzles, each with four reversing buckets in one reversing chamber.

The turbine was tested as received (practically unused) and at several conditions of simulated progressive blade wear for steam consumption at several attainable brake horsepower loadings, and at several speeds. The power shaft was connected by a flexible coupling to a 150-horsepower electric dynamometer. Steam was supplied at a line pressure of about 250 pounds per square inch gauge. During all tests, including the nozzle calibration runs, the steam was supplied from a superheater with a few degrees superheat after the throttle valve. This was in order to provide controllable and constant condition of steam during all tests so that all runs should be strictly comparable rather than endeavoring to approximate dry and saturated condition. Exhaust was into a condenser, but atmospheric pressure was maintained to minimize either steam or air leakage.

Main runs were made at speeds of 1000, 1500, 2000, 2500 and 3000 R.P.M., and at power outputs of 10, 25, 50, 75 and 100 horsepower as obtainable with each speed. The first series as above was made with the turbine as received. Prior to these runs the turbine bearings were put in a good free running condition and reference measurements made, then the turbine was thoroughly run-in during the steam nozzle calibration runs.

After conference with a representative of the manufacturer, and in view of the photograph of a worn turbine rotor as shown in Figure 2, it was determined to simulate the wear there shown. The manufacturer states that this photograph was taken in 1923 after twelve years' service of the turbine on a gas blower in the plant of the Lynn Gas and Electric Company. The turbine was operated continuously for that length of time, two minutes on load and two minutes idling, giving very wet steam to the turbine. The speed was 1800–1900 R.P.M., full load 190 horsepower, steam pressure 110 pounds per square inch.

To mechanically reproduce the wear shown in Figure 2, a small shaper was bolted to the turbine casing and suitable stops and tools so arranged that a cut could be made on the upper face of each blade which would be identical from blade to blade. The ragged wire edge left by the cutting tool was removed by hand

filing, but no particular effort was made to sharpen the edge. Nevertheless there resulted a more knife-like edge than was possessed originally. The width of the cut was made from the left-hand shroud 7/8 inch to the right to copy the erosion seen on the actually worn wheel, where the steam issuing from the nozzles was split by the blades to go into the buckets. At the right-hand side the blades performed no such splitting function. The first cut was made so that the edge indentation was 1/16 inch; after the second cut the indentation was 3/16 inch; and after the third it was 5/16 inch. The fourth and last cut was made to extend the 5/16 inch indentation clear across the blade. Figures 3 and 4 show, respectively, the 1/16-inch cut and the final condition. Figure 5 was made, after careful measurements were taken, to show in section the cuts taken and the angular relations between the various elements. An elementary force diagram is superimposed on the lower drawing to show that the torque arm is unchanged by wear of the blades, the steam jet discharging directly into the buckets before or after wear. No cuts were made on the blades of the reversing chambers. Actually, wear would occur on these blades on the right-hand side, where steam is received from the rotor buckets.

Successive series of runs were made after each cut, conditions and points being as nearly as possible made identical with those of the as-received runs. Measurements were made toward the end of the test to ascertain that the turbine bearings remained in the same condition throughout, and the dynamometer was kept in absolutely trustworthy condition.

The results of the test are shown on Table I and Table II. The actual steam consumption figures as obtained for given power conditions and speeds on test were plotted and drawn as very large scale graphs. The figures given in Table I are taken from these curves. The percentages shown on Table II were computed from the figures on Table I, using the as-received "A" condition at a cited speed and power as 100 per cent base.

At lower speeds, more than about 1/16-inch edge cut reduced the efficiency of the turbine appreciably. At near rated speed even excessive cutting under the nozzle produced little change in the efficiency. At higher speed the efficiency was appreciably raised

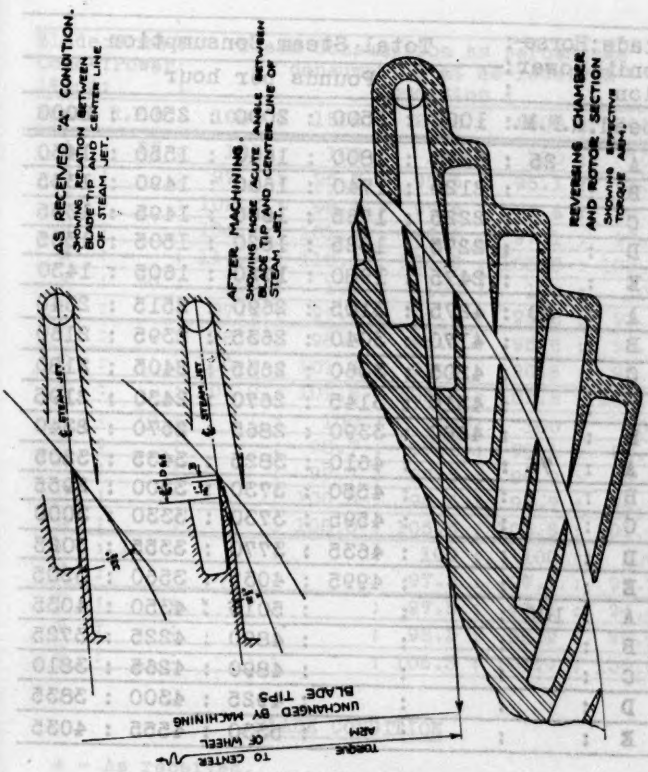
AS RECEIVED "A" CONDITION.
SHOWING RELATION BETWEEN BLADE TIP AND CENTER LINE OF STEAM JET.
AFTER MACHINING.
SHOWING MORE ACUTE ANGLE BETWEEN BLADE TIP AND CENTER LINE OF STEAM JET.
REVERSING CHAMBER AND ROTOR SECTION
SHOWING EFFECTIVE TORQUE ARM.
STEAM JET
STEAM JET
UNCHANGED BY MACHINING
BLADE TIPS
TORQUE ARM
TO CENTER OF WHEEL

FIGURE 5.—SECTIONAL ELEVATIONS OF TURBINE ELEMENTS.

TABLE I

Steam Consumption vs. Brake Horsepower

Blade Condition	Horsepower	Total Steam Consumption Pounds per hour				
Speed: R.P.M.		1000	1500	2000	2500	3000
A	25	2170	1800	1640	1550	1430
B		2125	1740	1600	1490	1335
C		2225	1765	1600	1495	1355
D		2275	1825	1620	1505	1365
E		2475	1980	1720	1605	1430
A	50	4275	3105	2690	2515	2340
B		4170	3040	2635	2395	2160
C		4205	3060	2635	2405	2180
D		4310	3145	2670	2430	2195
E		4660	3390	2865	2570	2340
A	75		4610	3825	3435	3205
B			4550	3730	3300	2955
C			4595	3730	3330	3000
D			4635	3770	3355	3025
E			4995	4035	3560	3205
A	100			5015	4350	4035
B				4890	4225	3725
C				4890	4265	3810
D				4925	4300	3835
E				5330	4555	4035

BLADE CONDITION

A - As received.
B - 1/16-inch deep x 7/8-inch wide cut.
C - 3/16-inch " " " " " "
D - 5/16-inch " " " " " "
E - 5/16-inch deep clear across blade cut.

TABLE II

Percentage Steam Consumption vs. B.H.P.

Blade Condition:	Horse-Power:	Steam Consumption as percentage of consumption at as-received Condition				
Speed R.P.M.:		1000	1500	2000	2500	3000
A	25	100	100	100	100	100
B		97.9	96.7	97.6	96.1	93.4
C		102.5	98.1	97.6	96.4	94.8
D		104.8	101.4	98.8	97.1	95.5
E		114.0	110.0	104.9	103.5	100.0
A	50	100	100	100	100	100
B		97.5	97.9	97.9	95.2	92.3
C		98.4	98.6	97.9	95.6	93.2
D		100.8	101.3	99.2	96.6	93.8
E		109.0	109.2	106.5	102.2	100.0
A	75		100	100	100	100
B			98.7	97.5	96.1	92.2
C			99.7	97.5	96.9	93.6
D			100.5	98.6	97.7	94.4
E			108.3	105.5	103.6	100.0
A	100			100	100	100
B				97.5	97.1	92.3
C				97.5	98.1	94.4
D				98.2	98.9	95.0
E				106.3	104.7	100.0

BLADE CONDITION

A - As received.
B - 1/16-inch deep x 7/8-inch wide cut.
C - 3/16 " " " " " "
D - 5/16 " " " " " "
E - 5/16-inch deep clear across blade cut.

by the cuts simulating normal wear. The increase in efficiency which resulted at all speeds and loads from the initial cut is attributed to the sharpening of the blades reducing wire-drawing and turbulence of entering steam. Positive identification of the cause of this increase was not considered a part of this test, and was not made.

SUMMARY AND CONCLUSIONS.

The turbine was tested for the effect of erosion of the blades. Actual wear was simulated by making shaper cuts on the face of the blades at that portion struck in operation by the incoming steam. The first cut, which indented the edges 1/16 inch from the original, corresponded to probably in excess of ten years' average operation. Successive 1/8-inch cuts were then made, and a final cut was made at the last or 5/16-inch indentation clear across the blades. Steam consumption was determined at various dynamometer loadings for several constant speed conditions for each condition of simulated wear.

At about normal speed of the turbine only the more excessive cuts produced more than about 2 per cent change in steam consumption. That the efficiency was increased at all test speeds by the 1/16-inch cut, and at some deeper cuts at higher speeds, is attributed to the knife-sharpening of the blades. It is probable that blade erosion of this type of turbines produces very little change in the steam consumption of an individual turbine during its normal life.

APPROXIMATE CALCULATION OF NATURAL TORSIONAL FREQUENCIES OF MULTIPLE GEARED MARINE OIL ENGINES.

By O. A. Banner, Member.*

For the ultimate success of oil engine reduction gears, it is essential that the torsional vibration problems of the complete drive are carefully considered because negative torques in the operating range due to torsional criticals must be avoided or the teeth of the gears will suffer.

The one-noded criticals of the system: Engine (or Engines) — Gear Unit — Propeller — can, with the exception of some quite special cases, be arranged below any sustained operating speed. Also, their range can be kept narrow enough to avoid trouble. Even quite frequent passing through this critical range at the low frequencies obtaining does not harm and leaves no traces on the gear teeth.

The severe criticals of the two-noded vibration must be placed so high that negative torques are avoided with the propeller racing. Good judgment should be used about the location of the more important minor criticals.

It is customary, before figuring the natural frequencies of the complete system, to make an approximate check using the two- and three-mass formulae. In quite a few cases, the results thus obtained are sufficient to make it unnecessary to figure with all the masses.

The peculiar conditions of multiple geared oil engine drives require certain caution in the use of the simple three-mass formula which shall be explained in the following.

The present investigation is confined to the case of the natural free vibration frequencies exclusively. The general vibration problem is not considered.

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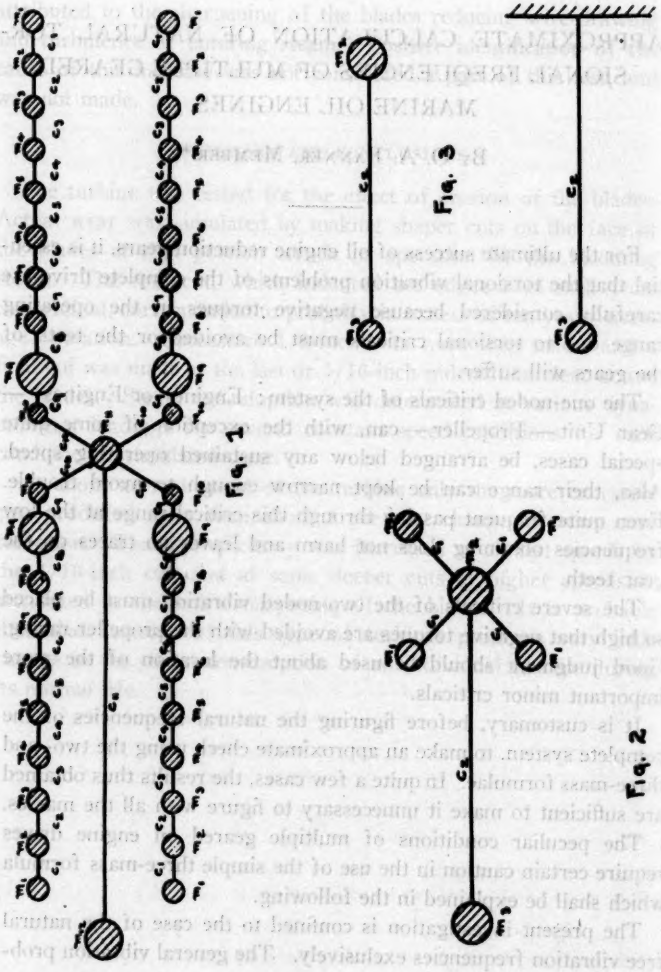


Fig. 1

Fig. 2

Fig. 3

Figure 1 shows the arrangement of a quadruple engine drive prepared for the calculation in the well-known way. All masses are reduced to common radius and common speed, all elastic lengths to common shaft diameter and common speed.

Couplings of the Falk flexible type have been assumed between flywheels and pinions.

m_1 to m_8	denote	cylinder masses
m_9	"	flywheel mass plus coupling half mass
m_{10}	"	coupling half mass
m_{11}	"	combined mass of the Gears
m_{12}	"	Propeller mass
c_1 to c_{11}	"	the shaft constants

It is assumed that the masses m_1 to m_{10} and shaft constants c_1 to c_{10} are, respectively, alike for the four engines.

If u_1 to u_{12} are the vibration amplitudes at any time of the masses m_1 to m_{12}, respectively, then the following equations for the equilibrium between inertia forces and shaft reactions exist:

For each engine —

$$(1) \qquad m_1 \frac{d^2u_1}{dt^2} + c_1(u_1 - u_2) = 0$$

$$(2) \qquad m_2 \frac{d^2u_2}{dt^2} + c_1(u_2 - u_1) + c_2(u_2 - u_3) = 0$$

. . . . . .

$$(10) \qquad m_{10} \frac{d^2u_{10}}{dt^2} + c_9(u_{10} - u_9) + c_{10}(u_{10} - u_{11}) = 0$$

For the gear unit:

$$(11) \qquad m_{11} \frac{d^2u_{11}}{dt^2} + 4\,c_{10}(u_{11} - u_{10}) + c_{11}(u_{11} - u_{12}) = 0$$

For the propeller:

$$(12) \qquad m_{12} \frac{d^2u_{12}}{dt^2} + c_{11}(u_{12} - u_{11}) = 0$$

This system of simultaneous differential equations is reduced by the substitution—

$$u_1 = a_1 \sin \omega t$$

$$\vdots$$

$$u_{12} = a_{12} \sin \omega t$$

where a_1 to a_{12} the maximum amplitudes and ω any of the frequencies of the different modes of vibration.

Carrying out the substitution, we obtain—

$$(1^1) \quad (c_1 - m_1 \omega^2)\, a_1 - c_1 a_2 = 0$$

$$(2^1) \quad (c_1 + c_2 - m_2 \omega^2)\, a_2 - c_1 a_1 - c_2 a_3 = 0$$

$$\vdots$$

$$(10^1) \quad (c_9 + c_{10} - m_{10} \omega^2)\, a_{10} - c_9 a_9 - c_{10} a_{11} = 0$$

$$(11^1) \quad (4\, c_{10} + c_{11} - m_{11} \omega^2)\, a_{11} - 4\, c_{10} a_{10} - c_{11} a_{12} = 0$$

$$(12^1) \quad (c_{11} - m_{12} \omega^2)\, a_{12} - c_{11} a_{11} = 0$$

Altogether we have four sets of equations, 1^1 to 10^1, one for each engine. These forty equations can be reduced to ten by adding the corresponding ones and eliminating the common factor 4. The correctness of this procedure can be shown as follows:

Take equation (1) for any two engines. As c_1 and m_1 are alike in both cases and ω is the frequency common to the complete system, it follows that the amplitudes a_1 and a_2 must be alike for the corresponding masses in order to satisfy the equation for any value of c_1, m_1 and ω. Similar deductions can be made for any corresponding set of equations.

Thus the system of equations 1^1–12^1 is sufficient for the calculation of the frequencies and the modes of vibration of the complete system which can be carried out using any one of the established methods. Any number of interchangeable engines connected to the gear unit can be considered by substituting their number for the factor 4, occurring in equation 11^1.

It is clear from the preceding that corresponding masses of any of the engines vibrate with the same phase and amplitude.

Reducing the problem now to a three-mass system with n engines connected to the gear unit, we combine all cylinder masses in the center of the crankshaft and the flywheel, coupling and gear masses in the flywheel.

Figure 2 shows this system.

Writing down the differential equations and using the same substitution as shown above, we obtain—

$$(13) \qquad (c_1 - m_1 a_1 \omega^2)\, a_1 - c_1 a_1 = 0$$

$$(14) \qquad (nc_1 + c_2 - m_2 \omega^2)\, a_2 - nc_1 a_1 - c_2 a_3 = 0$$

$$(15) \qquad (c_2 - m_3 \omega^2)\, a_3 - c_2 a_2 = 0$$

The elimination of the amplitudes a_1, a_2, a_3 leads to the formula—

$$(16) \quad (nc_1 + c_2 - m_2 \omega^2) - \frac{nc_1^2}{c_1 - m_1 \omega^2} - \frac{c_2^2}{c_2 - m_3 \omega^2} = 0$$

The two values of ω which satisfy this equation are the frequencies of the two possible modes of vibration. They can be found by assuming a series of consecutive values for ω.

Equation 16 can be further reduced to—

$$(17) \quad \omega^4 - \omega^2 \left(c_1 \frac{n\, m_1 + m_2}{m_1\, m_2} + c_2 \frac{m_2 + m_3}{m_1\, m_3} \right) + c_1\, c_2 \frac{n\, m_1 + m_2 + m_3}{m_1\, m_2\, m_3} = 0$$

For $n = 1$ this equation becomes the well-known three-mass formula.

It is obvious that equation 17, so far as the two-noded vibration is concerned, must give wrong results in case more than one engine is considered connected to the gear unit.

The two-mass formula is derived from 17 by introducing $c_1 = 0$. We obtain—

$$(18) \qquad \omega^2 = c_2 \frac{m_2 + m_3}{m_2\, m_3}$$

Figure 3 shows the system.

From 18, the one-mass formula is derived by the substitution

$$m_2 = \infty$$

We obtain

$$\omega^2 = \frac{c_2}{m_3} \tag{19}$$

Figure 4 shows the system.

In order to judge the degree of accuracy of the different formulae, the actual case of a quadruple eight-cylinder engine arrangement has been figured. The relative values of m and c are as follows:

m_1 to m_8	= 1.0	c_1 to c_7	= 1.0
m_9	= 44.4	c_8	= 1.032
m_{10}	= 2.675	c_9	= 0.889
m_{11}	= 5.125	c_{10}	= 2.668
m_{12}	= 16.15	c_{11}	= 0.00573

The engine is located aft in the ship. The results are given in Table 1.

TABLE 1.

The exact values obtained by figuring with the complete 12-mass system show negligible variation of the one- and two-noded vibration frequencies whether one, two, three or four engines are connected. In the case of the two-noded vibrations, this was to be expected.

In the case of the one-noded vibrations, all approximate formulae, even 19, give close enough results.

Formulae 16 and 17 check perfectly in the case of one engine connected, but 17, as was to be expected, gives absolutely wrong results in the case of two, three or four engines connected.

Formula 16 gives in the present case nearly correct results for two engines connected.

In the present case, the flywheel vibrates against the engine masses, the node being between last cylinder mass and flywheel. But there are cases when it vibrates with the engine against the gears in the two-noded and against the engine in the three-noded vibration.

An idea of what may be expected can be obtained by figuring the two resonance frequencies of the system: gears — flywheel —

TABLE 1

VIBRATIONS PER MINUTE

Mode of Vibration		One-Noded				Two-Noded			
No. of Engines Connected		1	2	3	4	1	2	3	4
System	Formula								
1 Mass	19	159	159	159	159				
2 Masses	18	178.4	169.2	165.7	163.8				
3 Masses	17	177.6	170.0	166.2	164.6	1518	1079	879	762
3 Masses	16	178.6	176.6	175.4	173.3	1520	1615	1705	1808
12 Masses		167.8	167.3	167.2	167.1	1643	1653	1655	1657

one engine. In the present case the one-noded frequency of this system (the two-noded of the full system) will be found to be 1510 V.P.M., with the node between flywheel and engine.

In an actual case, the procedure would be to figure the one- and two-noded frequencies using formula 16 for successively one, two, three and four engines, then averaging the values obtained for each frequency. As soon as desirable conditions have been sufficiently established, the calculation with the complete system may be carried out.

AIR CRAFT CONSTRUCTION IN THE CHINESE NAVY.

By Commander Bryson Bruce, U. S. Navy, Member.

The Chinese Navy, in the midst of the country's troubles, has laid the foundation for an air force. After time spent abroad in study and training, three young Naval Officers returned to China in 1918 and began to study the manufacture of seaplanes from the point of view of their own country. The Naval Air Establishment was started at Foochow at this time. Naturally, one of the first steps was to test native materials which might prove useful in the manufacture of the planes. These tests have been carried out, suitable native materials found, and several types of seaplanes built. Building has not taken place on a large scale; however, the present establishment could be very easily expanded to manufacture planes in large numbers. Additional workmen would need to be trained which would present no difficulty as there is plenty of potential material. The Chinese are expert at the kind of work required and have the infinite patience necessary to accomplish the most tedious task.

Foochow is located in the province of Fukien on the southern coast of China opposite the northern end of the island of Formosa. The Naval Air Establishment was located here for a number of reasons, among them being that there already existed at Foochow one of the principal Chinese navy yards with its facilities available.

The results of investigation of materials with information on these materials together with a description of the various types of seaplanes so far manufactured have been made available to the writer. One floating hangar has also been built for use on the Yangtse river, which is of original design. Undoubtedly now that China appears to be settling down to a stable form of government, much more time, energy, thought and funds will be spent on constructive work and the nucleus air establishment will be expanded to take care of the needs of the country in this regard.

Three young Naval Constructors are engaged in the design of planes and in charge of their manufacture at Foochow and have been since the establishment was founded. Each of these officers has had Naval training at home and extensive technical training and aeronautical experience abroad in American and England. Commander Yu Tsao Barr, the senior of the Naval Air Establishment, Foochow, graduated from Nanking Naval College in 1909, followed by practical experience in Vickers and graduation from Durham University (Armstrong College), England, in Mechanical Engineering.

In 1915, during the summer, he received flying training at the Curtiss Flying School, Buffalo, and in 1916 graduated in Aeronautical Engineering from the Massachusetts Institute of Technology, followed by a period in the Curtiss Aeroplane and Motor Co., at Buffalo, as an Aeronautical Engineer. With this experience, he returned to China and since 1918 has been at the Chinese Naval Air Establishment at Foochow. Commander Barr was known abroad as Tsao Yu. He is a member of the Society of Naval Engineers, China, Science Society of China and Society of Automotive Engineers, New York.

Commander T. Wong, to whom the author is indebted for the technical information contained in this article, has had similar experience to that of Commander Barr. He graduated from the Naval College at Chefoo, prior to going to England to attend Durham University and for practical experience in shipyards. He likewise graduated from M. I. T., and received flying training at the Curtiss School. In 1916-17 he was an Aeronautical Engineer with Pacific Aero Products Co., now the Boeing Airplane Co., Seattle. Since 1918 he has been engaged on his present duties. He is a member of the Society of Naval Engineers, China, Science Society of China, Chinese Engineering Society, and Associate Fellow of the Royal Aeronautical Society and the Institution of Aeronautical Engineers, London.

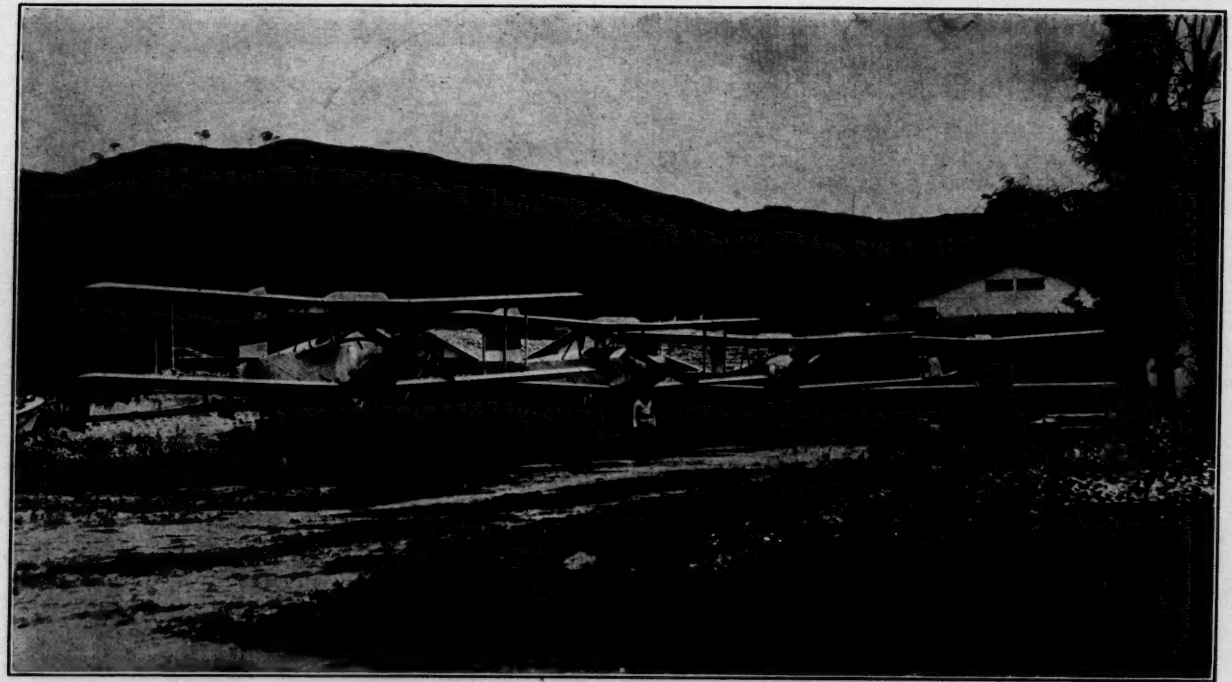
The third of this group, Commander Yee King Tseng, also graduated at the Chefoo Naval College prior to going to England for study at Durham University and for practical work in shipyards. He graduated from Durham in 1915 with Commander Wong in Mechanical Engineering. Commander Tseng has spe-

cialized in engines—Diesels at the Electric Boat Co., New London, in 1915-16, and aeroplane engines at the Curtiss plant in 1916-17. He is in charge of this work at the Foochow plant. He is a member of the Society of Naval Engineers, China, and Science Society of China.

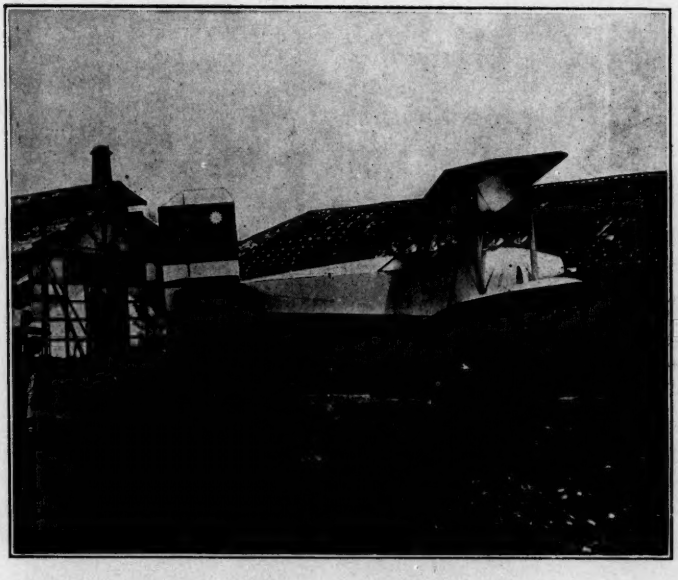
In order to save time and initial cost in the erection of new plant and installation of new equipment suitable for the work contemplated, a part of Foochow Navy Yard, including offices and workshops with some general metal-working and woodworking machinery, was allotted to the Air Establishment for a start. Machines, machine tools and factory buildings special to the construction of air craft were added from time to time as required. Such jobs as casting, electroplating, galvanizing, etc., which did not justify installation of plants on account of the comparatively small quantity of work to be done, were handled by the Navy Yard facilities. General office personnel, draftsmen and workmen were at first selected and drawn from the Navy Yard also.

At the same time that the Air Establishment was founded, a graduate course of Aeronautical Engineering was started in the Foochow College of Naval Construction, the officers of Naval Air Establishment acting as instructors. This consisted of both theoretical and practical work, the practical work being given in the workshops of the Air Establishment. Eventually nineteen graduated from this class and later on filled up practically all the minor technical positions in the Air Establishment. When constructional work was begun a number of apprentices were also recruited and trained.

The Chinese army had experimented with balloons in the last days of the Dowager Empress and founded a flying school and the nucleus of an Air Service as early as 1910; however, almost all the aeroplanes then in use were imported. China up to 1917 had little or no data or experience in the construction of modern aircraft. Therefore the engineers in the Air Establishment were confronted at the outset with a lack of scientific information on such home products as were of likely value in aeroplane construction. As it was the intention of the officers to use such home products as possible, the first thing to do was to test and carry out experimental and research work in order to discover materials



SEVERAL TYPES OF SEAPLANES DESIGNED AND CONSTRUCTED BY THE CHINESE NAVAL AIR ESTABLISHMENT, FOOCHOW, CHINA.



CHINESE NAVY SEAPLANE, TYPE "WU," BRISTOL "LUCIFER" ENGINE.

which would have the same or equivalent qualities as those that had been found to be suitable or in common use in the aeronautical industry abroad. In doing this, the question of supply was always kept in mind. It was desirable that the materials should be easily procurable in the local market.

TIMBERS.

Some ten species of soft and hard wood were tested according to the best scientific standards. Mechanical tests were carried out on a horizontal type of testing machine of French make and other tests such as specific gravity, moisture content, etc., were carried out in the physical and chemical laboratories of the Navy Yard.

The results of exhaustive tests showed that Shanmu, known also as Foochow pine, "China Fir" and "Foochow Pole," which abounds in the mountain forests of Fukien, possessed qualities equal to those of "Grade A" spruce as specified by the British Engineering Standards Association. This timber has since been used exclusively in the construction of main strength members such as spars, longerons, etc. It is also found in the provinces of Kiangsi, Hunan, Hupeh, Szechuen, Yunnan, Kwangtung and Chekiang.

Shanmu is found all over the above mentioned provinces from sea level up to about 7000 feet altitude, but does not occur where the winters are severe. It grows to a maximum height of over 100 feet and a diameter of over 2 feet.

It has a straight massive stem and a pyramidal crown. The branches are numerous, slender, short and horizontally spreading. The leaves, which are evergreen, are very narrow and stiff and with stiff, sharp needle points. They are usually dark above and more or less glaucescent underneath.

This tree is not grown from seed but is planted in the following manner: a tree is, in the first year, stripped of its branches and it then sends forth numerous little branches where the others had been cut off. In the beginning of the second year these are about 15 inches long, they are cut off with a slanting stroke and set out at a depth of 8 inches, with the face of the cut on the uphill side. They are planted in very thick rows and are thinned out year by year.

From Foochow alone the annual export trade in this timber amounts to a minimum of 8 million silver dollars. A maximum trade of 13 million silver dollars has been reported.

Shanmu is not a true pine and in some ways is not a true spruce, that is, it is a form which combines some spruce and some fir characteristics. It is not believed to be found in America.

Indigenous elm was found to have properties similar to rock-elm but workable lengths above 16 feet are hard to obtain. For floats and flying boat hulls, this offered no difficulty in the making of timbers, but for keels and chines sometimes splicing has to be resorted to. Except in size, Chinese elm is similar to American elm. It is found in the provinces of Kiangsi, Kiangsu and Chekiang.

Camphor wood, and especially its root, on account of its toughness, fineness, crookedness of grain and great strength, has proved most valuable in the making of knees or small bracing members and gusset plates in the construction of floats, hulls, and Warren girder type of fuselages. This wood weighs about 40 pounds per cubic foot and is a special product of South China. It is found in the provinces of Kiangsi, Kiangsu, Chekiang, Fukien, Kwangtung and rarely in Szechuan and Hupeh.

An old tree of camphor is one of the most majestic and graceful of Chinese trees. Trees often reach 90 feet in height and occasionally are found much over 100 feet high and 10 or more feet in diameter. The long and almost horizontal lower spreading branches often extend to tremendous lengths and one tree may cover a great deal of ground. The beautiful foliage, which is evergreen, is an added attraction, being bright green, smooth and polished or glossy above and pale green or with a glaucous bloom below. There are two kinds of camphor wood — red and white. The young leaves of the tree, which produces red wood, are pinkish and give the whole tree a wonderful glow that is beautiful to behold. The trees blossom small, pale yellow flowers in the early summer and bear seeds of the size of green peas but yellow in color.

It is a hardwood and grows about one inch in girth per year, so that a tree has to be fairly old before it can be profitably felled.

In Fukien these trees are found in great quantities in the mountain districts bordering on Kiangsi. Owing to its great size and difficulty of transportation, this timber is practically all brought down to Foochow in the form of boards. In the forests where modern machinery is non-existing, it is said that a large tree would take ten men as long as six months to saw into planks suitable for shipping. The boards are spicy and aromatic and they give a strong scent which is very pleasant to human noses but evidently not so to those of moths. This point is taken advantage of and the boards are generally used and prized for making moth proof clothes chests or linen cupboards. Roots of this tree, on account of their size, natural bends and good spike-holding power, go exclusively into the making of bulkheads and knees in " junk" or large wooden boat building.

Camphor is manufactured from this wood by distillation. A by-product is camphor-oil which is volatile and colorless. A tree, however, will not produce camphor until after it is about twenty years old, but even then it is not all trees that will do so. Expert workmen engaged in this line of work and backed by long experience, can always tell how much camphor a particular tree will yield. One thousand pounds of newly cut wood will yield about sixteen pounds of camphor and seven pounds of oil. This tree is not found in America.

" Baili" and " Nanmu" also on account of their toughness and fineness of grain, have been used on minor parts but these rather sparingly.

Baili, which is found in Northwestern Fukien, when translated literally means " white pear," but it is believed this is a local name in Fukien. For this reason and for the reason that it is not grown around Foochow, accurate knowledge of its characteristics are not available. A timber merchant reports that its leaves and flowers are rather similar to common pear trees but that it does not bear fruit. Its maximum height is between ten and twenty feet and diameter up to ten feet. This information, however, must be accepted with reserve. It is certain, however, that this tree must be of a fairly large size because the size of the boards or planks marketed in Foochow is always 7 feet 2 inches long by 2 feet 2 inches wide. The baili wood is very fine-grained and whitish in color. It is not unlike maple.

There is a kind of pear tree which is found in different parts of Kiangsi and Fukien but it is not certain whether this is baili. Inquiries are still being made on this tree.

Nanmu is an evergreen tree and has long oval shaped leaves, small, pale, green flowers and small, dark purple seeds. It is in the same species as camphor tree, that is, its general characteristics are rather similar to the camphor tree. Its average height is between ten and twenty feet. Four feet wide boards of this timber are easily obtainable in the market. It is a fine-grained hardwood and is usually used in furniture making, but among the rich people in Central and North China it is considered the best thing in which to pack and bury themselves. A large coffin made of well-seasoned, perfect boards of nanmu often costs two or three thousand silver dollars. It is found in Western China, along the Yangtse, and in Chekiang, Kiangsi, Fukien and Kweichow.

AVERAGE RESULTS OF MECHANICAL TESTS ON CHINESE TIMBER.

	Weight per Cu. Ft. lbs.	Fibre Stress lbs. per sq. in.	Compressive Stress, lbs. per sq. in.		Shear Stress, lbs. per sq. in.	
			Parallel to grain	Perpendicular to grain	Parallel to grain	Perpendicular to grain
Shanmu	31.5	7170	5660	1270	402	400
Nanmu	46.4	8700	5840	2400	655	570
Camphor-Wood	40.0	5410	3710	3900	558	550
Chinese Elm	35.2	6260	3780	1850	724	720
Baili	39.5	10200	5770	8330	998	900

FABRIC.

Of the different kinds of silk, cotton and ramie cloth tested, Shangtung silk and Kiangsi ramie cloth were found up to the strength required. A fair quantity of these two kinds of fabric was manufactured by the mills under special instructions and used on seaplanes. Shangtung silk, however, has a tendency to lose its tautness under abnormally humid atmospheric conditions. As these

conditions are rather prevalent in the late spring and early summer in Foochow the use of this fabric on machines which chiefly operated locally was discontinued.

DOPE.

Climatic conditions also affected two kinds of dope which were made and used after a great deal of experimental and research work. These were later replaced by imported dopes which were known as P.D.N. 12, a pigmented nitro-cellulose dope and V. 84, an aluminum nitro-covering. Three coats of this dope and two coats of the nitro-covering proved very well-suited to the semi-tropical climate and never gave trouble.

VARNISH AND LACQUER.

For the protection of wood in internal work such as wing frames, fuselages, interiors of floats and hulls a varnish made with Tung oil as its chief gradient is employed with great satisfaction. This varnish exceeds all the requirements as laid down in specification 2×7 of the B.E.S.A. It might be mentioned here that the success or otherwise of this varnish is more of a question of heat treatment than blending. As a matter of fact, tests on samples of imported varnish supposedly manufactured to fulfill the B.E.S.A. specifications turned out rather disappointing. Its rate of drying was low and it webbed or wrinkled badly. To a large extent, this, no doubt, was affected by the peculiar climatic conditions existing.

Specially treated varieties of Foochow lacquer, which is world famous, have been successfully used for protection against the weather on all external wood-work, against fouling on float and hull bottoms and against corrosion on steel fittings. Apart from its engineering value, lacquer can be finished in any color scheme desired and gives a brilliant and perfectly smooth surface and is most pleasing to the eye. When properly treated and skillfully applied it is very light in weight and has absolutely no tendency to chip.

Raw lacquer or varnish is obtained from lacquer or varnish trees. These trees are chiefly found in Shensi, Kueichow, Kwangtung, Chekiang and Fukien. They are handsome, medium sized

trees, 30 to 60 feet tall (very rarely 100 feet) with large graceful compound leaves, which turn to scarlet or orange in autumn. The sapwood is white and the heart yellow. The flowers are small and yellow in color and the seeds are round and flat in shape.

The varnish is obtained by making incisions in the trunk; the substance flowing from the wound is collected in bamboo receptacles. It is at first white but soon changes to greenish-grey and finally black, if exposed to air and light. The collecting of varnish is done in the summer. A tree will yield lacquer when it is 5 to 6 years old. The yield per year per tree is not great.

The best raw lacquer used in Foochow comes from Shensi and the next best from Kueichow. Kwangtung and Fukien raw lacquers are rather poor in quality. Moreover, the production of raw lacquer in Fukien is too small to meet the demand of the market.

Chekiang lacquer is mostly used around Shanghai where it is commonly known as " Ningpo Varnish."

Raw lacquer as sold in Foochow is contained in air-tight bamboo or wooden vessels. Before using, the lacquer is first filtered to get rid of its impurities and then it is transferred into long shallow wooden troughs and exposed to sunlight. It is there continuously stirred. The lacquer when first taken out of its container is greyish white and on being filtered and exposed to light and stirred, it turns gradually chocolate color and finally black. After a certain period of exposure to light and stirring, it is ready for use and is then known as " treated raw lacquer." The treatment depends entirely on the experience of the worker who, while working, takes the temperature, strength of sunlight and other climatic conditions into consideration. So far the treatment has not yet been put on a strictly scientific basis, that is, it is not yet carried out with a number of scientific instruments and worked to a set of definite formulae.

Treated raw lacquer without further treatment can be applied to wood floors, cheap furniture, etc. It is one of the most indestructible varnishes known, but it is opaque black colored and does not give a perfectly smooth polish.

By adding a certain amount of treated tung oil, the varnish becomes reddish or light brown in color. The degree of lightness depends on the amount of tung oil added and on the primary coat

of yellow colored liquid usually used. It is this further treated varnish or lacquer that is mostly employed in furniture work. It gives a very bright and polished surface.

For works of art or the commonly known Foochow lacquer-ware, a different kind of treated tung oil is added to the treated raw lacquer, which makes it almost transparent. To this lacquer is added different pigments. For all refined work in lacquer, certain primary coats of varnish and wood filler are applied and a large amount of polishing or rubbing is done after each coat.

This work is really a matter of experience, time and patience. A piece of high class "coreless" lacquer-ware often takes as long as six months to complete and herein lies the reason for its high price.

According to the process of treatment received, lacquer is divided into two grades, "long" and "short." Long lacquer takes about 24 hours to dry and short lacquer only from 1 to 2 hours. A peculiar quality of lacquer is that it hardens in a moist atmosphere, but remains moist or sticky when the air is too dry or when exposed to the sun. For this reason lacquer work, if possible, is always done in a dark room which is kept at a certain humidity by having water sprinkled on the floor at frequent intervals.

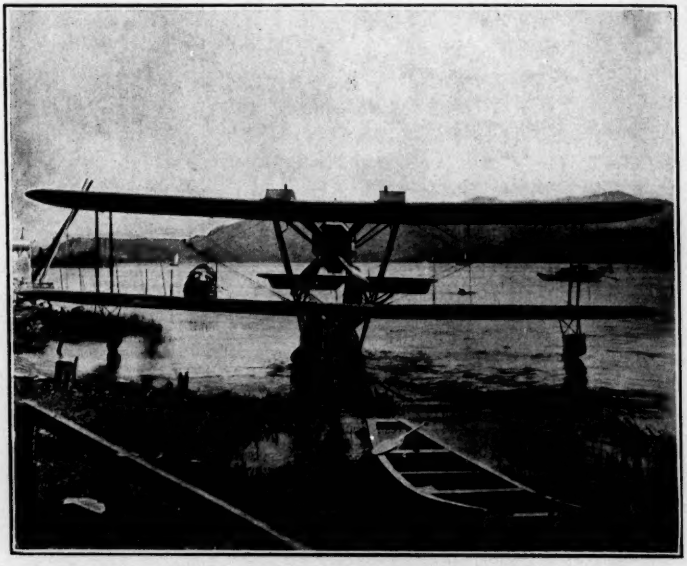
Raw lacquer is peculiarly poisonous, to some people more than to others. It causes eruptions of the skin, similar to poison oak. A person sensitive to it may be affected by its smell.

AIRCRAFT.

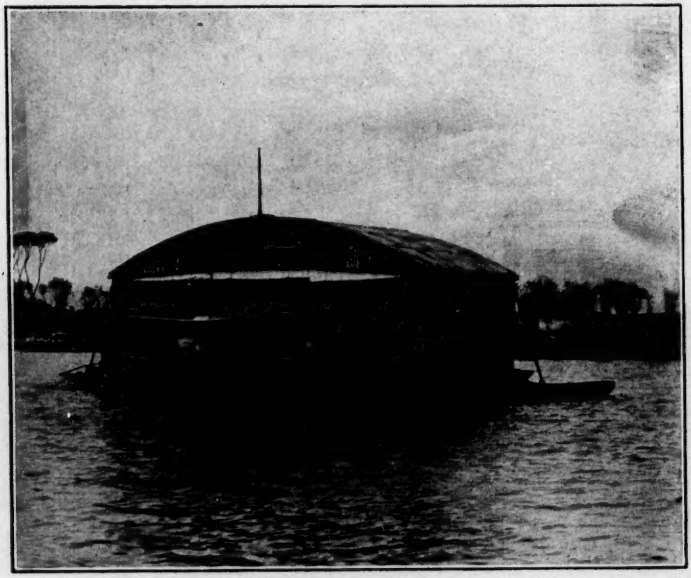
The first type of seaplane designed and constructed was for training purposes. The chief feature of this type was low landing speed, good stability and controllability and robustness of construction, especially in regard to the floats and float structure. These features were important because it was realized that a student under training would frequently, through inexperience, misjudgment or otherwise put a machine into awkward attitudes or make bad landings and if these were not well taken care of in the design and construction there would be large casualty lists.

Type "Char," as it was styled, was a tractor biplane with twin-float arrangement. The wings were of Shanmu and silk construction with I-section spars and built up ribs, and inter-plane bracing

was secured by two pairs of streamline Shanmu struts on each side of the fuselage, together with high tensile steel bracing cables, these were double for flying wires and single for landing wires. Wooden fillers were inserted in the spaces between all the double bracing cables. This had been found by the designers from wind tunnel tests in 1916 to reduce the resistance to a value much less than that of a single cable of the same diameter. The top plane, which was larger in span than the bottom one, had a dihedral angle of 3 degrees and was supported above the fuselage by the usual arrangement of center section struts and bracing cables. It was in three sections, two outer ones attached to a small center section. The bottom plane was straight and made up of two sections. The ailerons were mounted on the top planes only. The tail surfaces were of similar construction to that employed for the main planes. Two seats in tandem were provided in the fuselage which was of the orthodox wire braced rectangular girder construction, the longerons and struts being of Shanmu and the bracing steel wires. The floats were of streamline form in plan, ending in a transom stern. Their sides had a slight tumble-home and decks a slight dome. They had V-shaped bottoms with single steps which were ventilated. Keels and chines were of elm and frames of Shanmu. Planking was by two plies of narrow strips of Shanmu which had bevel edges to allow for shrinkage and expansion which sometimes occurred to some extent in service. The two plies were laid at right angles to each other, each ply being at about 45 degrees fore and aft. Between the two plies of planking there was a layer of fabric impregnated with marine glue to ensure watertightness and strength. Each float was divided into five water-tight compartments, each compartment being provided with a hand hole or inspection hole on deck. The inspection holes were made of aluminum casting and their covers were screwed on, the whole being flush with the deck. Watertight bulkheads were fitted at the joints of the under-carriage struts and also at the step. In this manner the landing forces and the load on beaching were transmitted through the bulkheads to all the strength members of the float. Cleats for towing, mooring, etc., were fitted at the bows of the floats.



CHINESE NAVY FLYING BOAT, 360 H.P. ROLLS-ROYCE ENGINE.



CHINESE NAVY FLOATING HANGAR FOR USE IN YANGTSE RIVER.

For transportation between hangars and slipways steel trucks or trolleys with suitable wood and felt cushions for the floats were provided for each machine. Each trolley was fitted with four rubber-tired wheels, the axles of these were all in one line and located very slightly forward of the center of gravity of the machine. Small wheels were fitted on the ends to prevent damage of the machine due to accidental pitching of the trolley while on the ground or slipway.

This type was powered with Curtiss OXX-5 engines developing 100 H.P. at 1400 R.P.M. Two-bladed wooden propellers were employed and radiators were mounted on the noses of the machines.

TYPE "YEE."

In general design it was similar to type "Char" except that the top and bottom spans were made equal and the wing area smaller. Areas of control surfaces were also reduced since the main plane area was. The main construction was substantially the same as type "Char." The nose of the fuselage was modified to accommodate the Hall Scott A7-a engine with which this type of machine was powered. It developed 100 H.P. at 1400 R.P.M. Side radiators were employed, one mounted on each side of the fuselage and slightly behind and above the engine.

Both types "Char" and "Yee" were designed to a minimum load factor of 8 and as their loaded weights were about the same, 2100 pounds. "Yee" was the faster of the two since its wing area was smaller. Both these types had a fuel capacity for 3 hours flying near the sea level and, as at first constructed, the gasoline tanks were installed under the front or student's seat. Feed to the engines was therefore by pressure. On later machines the tanks were mounted inside of the top center sections which were designed to a good thick wing section. Feed was hence by gravity and direct, thereby not only weight was saved but, most important of all, constant troubles incidental to the pressure system were eliminated. Thereafter gravity feed was adopted on all types of machines.

TYPE "BEENG."

This was a single engined flying boat designed for military purposes and, if desired, with slight modifications could be adapted to passenger carrying.

It was a biplane structure, the top and bottom planes of which were equal in span and each was made up of three sections, one center section and two end ones. The bottom center section was attached to the boat hull by short steel tubular struts, the lower ends of which were fitted on the hull just above the waist line.

The engine supporting structure of steel tubular struts and bracing cables was mounted on top of the bottom center section and above this and also on a structure of tubular struts and bracing cables was mounted the top center section. On to the top and bottom center sections were attached the top and bottom end planes, the interplane bracing of which was by means of a single pair of tubular struts with the necessary bracing cables on each side.

The tail surfaces were mounted on the stern of the hull. The fin and rudder were of symmetrical section while the tail plane and elevators were of an inverted wing section, as practically under all attitudes of flying a down load was required. A trimming gear was fitted on the tail plane which was adjustable from the pilot's cockpit during flight.

Spars on the main planes and horizontal tail surfaces were of box section the flanges of which were made of Shanmu and webs and walls were of Shanmu and camphor wood. Spars on the fin and rudder were Shanmu I-sections. All ribs were built up of Shanmu.

All exposed tubular struts were faired up by hollow wooden tail pieces to a good streamline form and finished up by a wrapping of fabric and lacquered.

The hull which was 36 feet in length and which was of the flexible or Linton Hope type of construction, was in plan of streamline form and in section circular on top and concave V-shaped at the bottom.

The keel and keelson were of elm and were continuous from stem to stern. A large number of fore and aft stringers of Shanmu were evenly distributed around the periphery of the hull.

Small timbers of elm, closely pitched, were bent round these stringers, ending at the keel. Floors of elm spaced at varying intervals ran around the lower half of the bottom and were continuous through the keelson. The spacing was very close at the main step where the load on water was greatest and about half as close in the forebody and tail.

At intervals of a few feet continuous single hoops of elm were fastened to the top of the keelson and stringers. Where the bottom center section struts were fitted, double hoops of elm, one on each side of the stringers were fitted as above.

To the timbers was copper riveted the planking which consisted of two plies of narrow width of Shanmu wrapped right around the hull, ending at the keel. The two plies ran roughly at right angles to each other. In between the two plies, fabric and marine glue were used similar to the float construction.

The planing surface or bottom which was continuous from the stem to the aft or second step, was attached to the bottom of the main hull through a series of stringers and hoops rather similar to the main hull construction. This bottom was kept clear of the hull at practically all points, thus forming with the hull and fin tops a complete double bottom. A brass protected false keel was then fixed on, sealing up all the ends of the planking. Finally the boat was completely covered with fabric and lacquered, making an exceedingly watertight and strong job.

For lateral stability on the water, a pair of wing tip floats was fitted under the interplane struts.

The engine, which was a 360-horsepower Rolls-Royce, was mounted between the wings and drove a 4-blade tractor propeller, the tips of which were a few inches in front of the leading edges of the wings and above the deck.

Some little distance in front of the propeller was the pilot's cockpit and forward of this was the first gun cockpit. A second gun cockpit was situated near the trailing edge of the bottom center section. As gasoline was contained in tanks on the top center section, the hull space between the pilot and aft gun cockpits was free from the stowage of ammunition or if for passenger carrying, four seats could be easily arranged.

On later boats of this type, the horizontal tail was altered to a biplane structure. The bottom elevators were reserved for trimming purposes and the top ones for ordinary control.

TYPE "WU."

This was a general purpose machine of 120-130 horsepower. It was a straight-forward twin float single bay biplane. Its chief features were simplicity of construction and ease of maintenance. The wings and tail surfaces followed the usual construction of the Establishment. As no great speed was aimed at, the floats were designed rectangular in plan form, decks and sides flat for ease of production and as the V-bottoms of the floats were so shaped that no great sheets of spray were thrown up to the bottom planes, ailerons were fitted on the bottom planes only, thereby the aileron control was appreciably simplified.

A feature different in this type from previous types was its fuselage construction. In this, except for the engine mounting in the nose, no metal fittings or wires were employed. It was built up in forms of Warren girders, which consisted of four longerons of Shanmu a number of struts of the same material. The attachment of the struts to the longerons was by means of two-ply camphorwood gussets glued and copper riveted. Gusset plates were applied on both sides of the longeron and struts for strength, and for this purpose, the strut joints on two adjoining sides were slightly staggered. For torsional rigidity, camphorwood knee braces of natural curved grains were fitted on all corners. This construction proved very rigid and strong under bending and torsional tests and in actual service was entirely satisfactory. Its advantage over the ordinary braced type of monocoque type must be obvious since no great numbers of fittings, wires and turnbuckles or special forms and jigs were required and no constant truing up was necessary in service.

This type of machine was equipped with Bristol "Lucifer" air-cooled engines.

TYPE "DING."

At present the Naval Air Establishment has near completion some torpedo and bombing planes. It is regretted that no details of these machines are available for publication at this time.

FLOATING HANGAR.

In addition to the different types of seaplanes that have been designed and built, a floating hangar was designed and constructed for use of a seaplane.

It is built on a hull which consists of 2 longitudinal steel pontoons connected together by transverse girders and deck plating. Each pontoon is divided into 5 watertight compartments; the center one is the flooding compartment and the other 4 are holds and chain lockers. In the center of the hull there is a trough in which the floats of the seaplane normally rest. At the front end of the trough there is a hinged flap which acts as a watertight door.

When the seaplane is not in use, the hull is at its normal waterline, the flap door is closed and the bottom of the trough is dry. When operating the seaplane, water is pumped into the flooding compartments and the hull is sunk to its low waterline; the flap door is then opened and the machine is water borne. It can therefore be taxied out. Sinking or floating of the hanger is accomplished by means of a centrifugal pump driven by a gasoline engine.

The overall dimensions of the floating hangar are:

Height, feet and inches	26-6
Length, feet and inches	70-8
Width, feet and inches	35-4
Normal draft, feet and inches	2-10½

This hanger was designed for use on the great Yangtse River. Its chief advantages are its mobility, as it can be towed anywhere, and its ability for operating in shallow waters. It was designed by the officers of the Naval Air Establishment, Foochow, and built in 1922 by the Kiangnan Dock and Engineering Works.

The Chinese navy for economical reasons usually sent their cadets to the army flying school in Peking or to civilian flying schools abroad for training on land machines or recruited aviators who had been trained elsewhere. On joining the navy, they were assigned seaplanes for practice and when proficient were given permanent commissions as aviation officers. Very little actual training on a large scale has been done up to the present.

Comparatively few accidents in flying have been experienced by the Chinese navy and none attributable to failures of materials. There has been only one fatality. All accidents resulting in damage to planes have been due to the inexperience of the student aviators.

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LATEST RESULTS FROM TUNGSTEN-CARBIDE TOOLS.
Machinery, June, 1929.

* EDITOR'S NOTE: "Propulsion of Ships by Modern Steam Machinery," a paper read before the Institute of Naval Architects, March 31, 1929, was published in the notes of the May, 1929, number of the JOURNAL. The comments herewith quoted refer to this paper.

WANTED: AN UNPRACTICAL EDUCATION.

By JOHN C. PARKER,* BROOKLYN, N. Y.

I have a son who is nearly 13 years old. You and I have a common problem with relation to that boy, I as the parent, you as possible teachers; or if a former teacher may momentarily revisit the pale glimpses of the moon, I shall identify in my own person both the teacher's viewpoint and that of the parent. Since prior to my seven-year period as a teacher I was an employer and have returned to that relationship, may I try to talk in all three capacities. Again, as a workman who has been somewhat educated may I talk as well from the point of view of that least considered victim in the process, the graduate himself.

I believe that the true interests of all parties of this process, the student, the parent, the employer, and the teacher—significant in the order named—are identical, that what is broadly good for one is to the best interest of all.

For this boy of mine, for his greatest usefulness in life and for his personal happiness, for the gratification of my hopes and ambitions for him, for what he can do in the world, serving the kind of employer I hope he will have and for the biggest satisfaction of the teachers under whom he may elect to study, I hope he can be given through the years of high school and college the sort of a thing that is commonly called an unpractical education.

I am basing this hope on the firm conviction that the kind of stuff that is usually understood as practical is actually most unpractical, and that the kind of thing that may be styled useless really has the highest utility.

There is a persistent demand from parents and employers, from trade associations and from donors, from the makers of curricula, and from the youngsters themselves whether in primary, secondary, or university institutions, that pupils be given something useful, meaning thereby, I suppose, something with which they can earn a living.

In so far as this is in protest against a snobbish classicism, I sympathize with it most fully. I have little patience with the idea that subject-matter loses its cultural or educational value merely because it is related to an honest world of affairs of physical fact and of live processes. On the contrary, a vital and vigorous education can come only through being related to life.

My complaint would be rather that training youth to earn a living is not education at all; second, that a specific training may keep the youngster from earning the best kind of a living; and third, that it can't be done in school anyhow.

I assume that it is not a legitimate function of educational institutions to take the funds of society as a whole to fit certain particular individuals to earn a living through advantages not universally enjoyed. On the other hand, I conceive the real purpose back of state support, back of the social support of education, to be the fitting of students to live richly and fully and to contribute most broadly to the welfare of the social group who have paid for their education.

Whether this be the true purpose of education or not for this boy of mine, I am considerably more interested in whether he shall be able wisely to spend and fully to enjoy what he earns than I am in the amount of those earnings or the ease with which they are acquired.

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* Vice-President in Charge of Engineering, Brooklyn Edison Co., Inc., Mem. A.S.M.E.
Address at a meeting of the Society for the Promotion of Engineering Education, Brooklyn, N. Y., December 8, 1928.

I would infinitely prefer that education fit him for happiness and decency in poverty than for wealth acquired through the sacrifice of himself and his character.

Grant that I am wrong in this. I know perfectly well that his young mind will absorb only a certain amount and that if particular training in a too narrowly defined art be given him, life may perpetrate one of its ghastly jokes in presenting narrow opportunity along that line and big opportunities in fields for which he has not been trained at all.

Any one who has lived in an actual world has seen literally thousands of cases in which men, young and old, have been trained in an art which has become obsolete and have seen the victims forced to degrade their social usefulness through lack of adaptability to changed conditions. We have seen cases in which the fields chosen by parents or advised by teachers or selected by the students have not in the sequel proved what they were thought to be, and we have seen men left standing in forward-moving organizations merely because they had not that breadth of fundamental training which would enable them to move with the current.

Granted, then, that just so much can be accomplished in sixteen or eighteen years of formal schooling, I want those things to be of such a character that my son can use them in whatsoever field of human endeavor may be suited to him and may offer itself, and if he chooses some particular range of studies I want to be sure that, aside from and superior to the subject-matter, those courses shall have a content which shall do things to him in the way of developing intellectual vigor, standards and ideals, and mental processes which can be brought to bear on any of the problems of life.

MATHEMATICS A SPUR TO IMAGINATION.

I hope he will be interested in pure mathematics, and I don't care whether he is as ignorant as his sire concerning the equation to or the shape of a lemniscate; that really doesn't matter after all and the world probably would have been quite as rich had the curve never been discovered, but if the youngster meets it I want him to be able to see some sheer physical beauty in the grace of the thing. I hope that his imagination will be arrested when he discovers that the shifting of the constants in the equation of a hyperbola creates a beautiful and moving series of exquisite curves all tied together at infinity, passing through two intersecting straight lines and filling voids in unoccupied quadrants. Here will be something that will arouse his imagination, that will teach him that order which is heaven's first law, which will enable him to project his mind beyond the immediate and the particular into a wondrous field of generalized relationships.

These values in his mathematical studies will be as great whether he follows the law, preaches the gospel, administers to the physical needs of the sick, or runs a merchandising establishment.

And if he elects to take a course in engineering I hope that he will bring out of it a knowledge of the conduct and behavior of materials and of the processes by which industrial human progress has taken place, which will enable him to understand a life which has its very foundations in the creative works of organized industrial society. I shan't, however, be very much concerned with whether he has been taught some particular form of valve gear or somebody's pet formula for buttoning up a riveted joint.

At this point let me say just one word about the real values in all these courses in pure and applied science. They deal or should deal with objective truth rather than with speculation or opinion. If bigly taught they will promote a higher type of imagination than ever could be had from subject-matter that concerns itself only with speculation. It is a finer vision that

can constrain itself to things proved and learn from them the great generalized truths of the universe, that can perceive the romance back of the lifting of a continent from the ocean bed—this is an infinitely finer type of vision than the hothouse variety that can find beauty and romance only in artificiality, be it never so exquisite.

But beyond all this, our scientific and professional courses have a tremendous advantage in their objective character. Being that, if they do no other thing, they should develop love of the truth, not moral integrity alone but intellectual candor, which in my mind is pretty close to the beginning and end of all the virtues. If you teachers will do some things of this sort for my son I know that he will approach life unafraid, that he won't try to fool himself into believing that his selfish desires are the will of Jehovah or that he is more important than his fellow-men.

If imagination and integrity can without preaching be developed in our professional courses, such education will be the more persuasive because it has been born of things to which it might not superficially have been suspected to be germane.

I have suggested that no school can give a narrowly practical training. Any one who knows both teaching and practice knows this full well. The life of the student is artificial—and should be. He is forced to march more or less in lockstep with his fellow-students through a more or less definitely-laid-out curriculum. Life doesn't work that way. Individuals in the ever-changing exigencies of a real world forge ahead or drop behind or move side by side up to a point of divergent influence. Decent or brutal competition exists. Some assume leadership—others follow. The whole atmosphere of school and of life is necessarily and essentially different.

Practical design concerns itself with intricately organized social and economic processes: the most trifling engineering device must be designed for production under the peculiar conditions and restraints of an actual factory, to be marketed through real and frail human channels, and to be used by actual ultimate consumers. Ninety per cent of the problems of a designer lie in the wise determination of these and other constraining facts, in the selection of the field which he shall endeavor to cover, and in the satisfaction of others that his determinations are sound. His work is primarily in the discovery and statement of his problem. No school can ever hope even remotely to assemble an environment of actuality about any problem—of necessity it must formalize an artificial problem if indeed it would avoid giving the student a most unpractical impression that his subsequent problems in actual life will come to him all neatly wrapped up in packages, labeled, and classified.

Grant even that real problems could be introduced within the school walls, the question yet remains as to what value attaches to the specific practical information that is gotten from solving them. Any professional work is a live thing, related to rapidly changing life. The ink will not have time to dry on the student's diploma before the quantitative facts, social emphasis, and the problems that are crying for solution shall have changed. What point, then, in teaching the youth formulas, data, and processes howsoever accurately if necessarily obsolete? This were a most unpractical practicality.

But, you urge, the world won't take our goods unless we purvey to its needs. Industry insists that we must turn out our graduates to its end. Maybe so—I can't ask you to take my personal wishes as an index of what the world at large wants, but for what they are worth let me state them here. As an administrator I am not particularly desirous of having young men come to us full of facts which would be slightly interesting if they were right, but which, in the main, will have to be unlearned because they just

aren't so. If a teacher can get hold of the facts of my industry in order to communicate them to his students he will have to get hold of those facts from the industry itself, and so, if you please, the student will be merely bringing back to us, a little late and a little shopworn, the specific facts that have arisen with us.

Put it the other way round. We shall get stuff thoroughly codified into a practice which can be carried out by routine workers long before it can be formulated into school instruction. We want something very much bigger than this. We need young men of keen perception, sound and orderly analytic ability, of sympathy and of understanding, to analyze the facts as they arise from day to day, to draw new conclusions from them, to generalize these conclusions into canons of the truth, and to devise new practicalities of putting them into effect. This sort of thing, in that it pays big dividends or increases the social value of our service, is practical with a vengeance, and it is the sort of thing for which every industry and every profession and every art is ahungered and athirst whether it recognizes the need or not.

TEACHING OF PRACTICE MAY BE THE TEACHING OF BAD OR OBSOLETE THINGS.

Let it be thoroughly understood, too, that the teaching of practice may be the teaching of obsolete or bad things. All practice contains some elements that are not worthy to be continued, and some practice is positively wrong. The major part of the life work of any man must be to create new practice to correct and to improve what exists.

The teaching of practice tends to the continuation of evils rather than to the type of mind that makes for betterments.

None of the world's great discoveries and inventions have come by the process of yielding up, like a squeezed sponge, these things which the mind has soaked up, but rather as the result of critical and creative and impatient mentalities.

What little of practical training may be necessary can very quickly be grasped by a young man fundamentally and thoroughly educated through his contacts in the only school of practice—life itself.

I am telling you of the need of my industry and of my profession as I see it, and telling you of what I personally would ask of the teaching profession. Is it a practical program of education to make education unpractical? Will the world accept the goods which the teacher is offering? My answer must be the very practical one of experience during seven years as an educational administrator. During the period to which I refer, the staff of which I was in charge endeavored solely to give a broad and fundamental education, concerning ourselves much more with what we did with our students than with what incidental facts they accumulated. This was done in the face of criticism from other teachers, complaint by the students, and insistent practical demands from industry. The parents happily left us alone. I say this program was adopted in the face of criticism: that was, initially. Enrolment dropped and gradually mounted considerably above what it had been before the elimination of every practical course offered by our department. We didn't at any stage in the game have any difficulty in finding places for our students with the people with whom we wanted them to connect; indeed, during the lean years of 1921 and 1922 the difficulty was rather to keep the youngsters from being inflated by too much solicitation on the part of employers who had had samples from previous crops. Six years later I find many of the boys who went through that kind of a process occupying rather unusually distinguished positions in industry, and am glad to say that I have been able and willing to use several of them in my own organization.

What industry will accept is altogether a question of having faith in the program and the courage to carry it through.

THE DUTY OF THE TEACHING PROFESSION.

And how for the teacher himself, what shall we say? Must he always listen to the cheap deprecation of his calling? Shall he concern himself with the stupidity of a world that damns with the label "pedagogue"? Is his way out to attempt to do something practical and useful, to go through the motions of practicality necessarily with his tongue in his cheek the while, or will he recognize that it is better to teach well those things that can be taught than to teach badly the things which have a glamorous glitter on the pages of a college announcement? Can he not—*must* he not—recognize that it is more important to educate and to develop men for life than it is to satisfy the whims and caprices of students or of possible employers or even to offer those courses which a president or a board of trustees ask of him as a cheap form of parlor trick? Again, may one be just a little practical in discussing this unpractical subject. One hears much of the poor pay that is given to the teaching profession. In part this is right and in part the just suffer with the unjust, but I rather suspect that were our schools to have more faith in education and to be less compliant in giving functional training they would deliver goods of a kind that the world really wants, and that the natural law of supply and demand would insure vastly more adequate compensation for teachers as teachers than it will give to school men who make a most inadequate pretense of being anything but what they are.

And so it is, gentlemen, that as the victim of a university education, as an employer, as a father, and as a teacher I am putting out my advertisement calling for an unpractical education in the full assurance that the right kind of unpractical education is the only kind that is most practical; and just in proportion to the shortness of the period that the individual can spend in school would I lay more stress on his getting from school those values which can be had nowhere else, so that the least favored shall have the assurance of a more nearly even start with those who can spend more time and perhaps in the graduate's school develop particularized training.

All of which is very unpractical unless one suggests an answer to the question—"What are you going to do about it?" Well, I have no faith that these remarks of mine are going to cause boards of trustees and curriculum committees to change their curricula over night, but I have a hope that those of you who have some faith in the theoretical aspects of education and who have somewhat of a pastoral interest in the boys committed to your charge will recognize that however practical the educational machine in which you are a cog, you can individually so direct the conduct of your courses that your students in later life may arise and call you blessed for having given them something useless—something on which to build for themselves.—"Mechanical Engineering," July, 1929.

DIESEL VS. STEAM—COMPARATIVE COSTS.

By Edward B. Pollister.*

The paper read by Mr. J. Johnson before the Institution of Naval Architects, in London, England, reveals a qualifying intimacy with steam practice, but so superficial a knowledge of motorships that the suggestion may be

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pardonable that Mr. Johnson might better have limited his paper to the titular subject, "The Propulsion of Ships by Modern Steam Machinery."

Logical investigation seeks the source of knowledge, as indicated by the Packard slogan, "Ask The Man Who Owns One." This is especially true of new types of machinery equipment. No one seriously interested would seek information on Diesel engines from steamship operators. Also, it is not surprising that motorship owners are reluctant to reveal the benefits of their adoption; nor the adherents of steam feel called upon publicly to justify the wisdom of their course.

Meanwhile, motorship owners are building three times as many high-powered motor liners as there are geared-turbine vessels under construction. In spite of the not unnatural antagonism of engineer steam adherents, of manufacturers of steam turbines, boilers and condensers, and of shipyards with steam but without Diesel building capacity or personnel, the total power of marine oil engines under construction throughout the world is (estimating turbines built in Germany) approximately equal to that of steam turbines and reciprocating engines combined:

LLOYDS RETURNS, MARCH 31, 1929.

Steam engines	573,508 indicated horsepower.
Turbines (excluding Germany)	484,600 shaft horsepower.
Oil engines	1,238,675 indicated horsepower.

This internal combustion encroachment on century-old marine steam power has taken place during the last ten years. On land, the small steam plant of up to 7500 horsepower has already been practically forced out of existence by the Diesel. It follows that the Diesel will replace small marine plants; and, on land, power plants of 15,000 horsepower would not be considered large. Mr. Johnson's paper should therefore be closely scrutinized before his conclusions are accepted, with reference to the predominant position of the motorship.

As reliability and reasonable upkeep cost are better established for marine Diesels than for new high-pressure steam propulsion, the two most vital factors in the comparison of Diesel vs. Steam, are:

Annual fuel cost of the motorship.
First cost of the motorship.

Mr. Johnson fails to establish either of these, with any reasonable degree of accuracy, authority or actual performance of motorships.

The rapid adoption of the marine Diesel is due largely to a gain over steam from well under 20 per cent to over 30 per cent thermal efficiency. This cannot be gainsaid. These efficiencies are accurately known; yet, Mr. Johnson presents more favorable fuel consumption figures for the *Duchess of Bedford*, but based on shaft horsepower output which cannot be accurately determined for a geared-turbine drive. Torsion meters are well known to be decidedly inaccurate. Mr. Johnson's figures show 16,000 shaft horsepower required to drive the *Duchess of Bedford* 17½ knots.

Citing Mr. Johnson's opening remarks, "that the economics of ship operation constitute the final court of appeal in these matters," consider the economics of the *Duchess of Bedford*—the *Viceroy of India*, which attained a speed of 18¼ knots under full load with turbo-electric power accurately measured of 13,322 brake horsepower (see "The Engineer" of March 15, 1929), and the motorship *Kungsholm:*

FROM THE BRITISH MOTORSHIP.

	Duchess of Bedford (High-pressure steam)	*Kungsholm* (Diesel)	*Viceroy of India* (Turbo-electric)
Gross tonnage	20,000	20,400	19,700
Length between perpendiculars, feet	580	580	612 (overall)
Molded beam, feet	75.3	78	80
Draft, feet	27	29	.....
Depth, feet	41.8	42.8	.....
Designed shaft horsepower	18,000 (16,000 in service)	15,000	16,000
Designed speed, knots	17½	17½	18¼

Compared with the daily fuel consumption of the *Duchess of Bedford* at 17½ knots for propulsion alone of 97 tons, the motorship *Kungsholm* made 18.3 knots with a propulsion consumption of 63.6 tons. This ratio is in line with known thermal efficiencies of Diesel and high-pressure steam plants, indicating that 16,000 shaft horsepower is too high for the *Duchess of Bedford*, and the equivalent fuel rate per shaft horsepower too low.

There appears to be a discrepancy between the comparative voyage fuel consumption records of the *Duchess of Bedford* and the *Montclare*, and the figures given in Table III for the same ships, which cause one to question the actual shaft horsepower output. Assuming that the sea time of the two vessels was inversely as their average speed, their specific fuel consumptions, for all purposes, 0.625 pound and 0.98 pound, as stated in Table III, and their voyage fuel consumptions respectively 1443 and 2182 tons as reported; either the shaft horsepower of the *Duchess of Bedford* must have been 14,100 (instead of 16,000); or that of the *Montclare* 14,200 (instead of 12,500). Also, how was the total fuel apportioned to the propelling machinery and other services, respectively?

Is it quite fair, in Table XI, to compare hoped-for fuel consumptions of steamships, with fuel consumptions of motorships that were apparently old in the history of the Diesel? Or, in Table XIII, to base the weight of bunker oil on the aforementioned hopes of steam machinery advocates, as against the actual performance of old-time Diesels? In this connection it is noted that, in Table XI, Mr. Johnson charges a 30,000 shaft horsepower Diesel installation with a total fuel consumption of 0.48 pound per shaft horsepower; whereas in Table XVI he figures 0.42 for propulsion in a similar installation ranging from 4000 to 10,000 shaft horsepower.

The motorship *Vulcania* of 24,000 gross tons and 19,230 shaft horsepower consumed on her first voyage 0.407 pound per shaft horsepower hour for propulsion, and 0.434 pound for all purposes, at an average speed of 18.57 knots.

It is quite evident that Mr. Johnson is comparing *actual* performance on his high-pressure steam installation with *assumed* motorship fuel consumption, substantially higher than actual motorship performance records. Then, although he concedes a life of twenty-five years for the ship, he determines the comparative cost of Diesel and steam fuel, using, principally, as a basis the present temporary, artificial price of bunker fuel on the Atlantic seaboard, with a cost ratio of 1 to 1.9 in favor of steam. The actual average

throughout the world is approximately 1 to 1.25 and the published bunkering prices of May, 1929, show a difference of only 15 cents per barrel at San Francisco and Seattle and only 5 shillings per metric ton at Hong Kong between boiler and Diesel fuel. Over a 25-year life of the ship, more normal higher fuel prices may be reasonably anticipated. Why not look ahead a few years?

Again, Mr. Johnson points out annual fuel consumption totals based on eleven 16-day voyages, equal to only 176 sea days; whereas, many passenger motorships are at sea 230 days and more, while many motorship freighters are at sea over 300 days per annum. On much more severe turn-around service from Vancouver to Australia, the passenger motorship *Aorangi* has averaged for the past four and one-half years about 250 days at sea per annum.

The effect of basing repair charges on the one hand and fixed charges on the other hand, on such a limited sea service, is obvious. Undoubtedly, Mr. Johnson's estimate for high-pressure steam installation of renewal every seven years of condenser tubes, superheater tubes, air-heater tubes and boiler fire-row tubes would have to be substantially increased for something like 40 per cent increase in sea service—if high-pressure steam plants could be used at all in such arduous service. In face of such comparisons, Mr. Johnson's apparent generosity in not charging motorship repairs and extra lubricating oil consumption into the account should not be overvalued. The remarks in "British Motorship" for May, 1929, of Mr. Visker, superintendent engineer of the Netherland Steamship Company, which operates motor passenger liners, "The cost of repairs on oil-engine ships is materially lower than on steamers, with any good class of engine," would apparently not apply to the new high-pressure steam marine plant.

Taking Mr. Johnson's figures, without attempting to correct the demonstrably erroneous fuel costs, note the effect on his comparison by increasing the sea days from 200 to 300 per annum:

	200 days	300 days
Extra cost of fuel for steam plant	£ 8800	£ 13200
Repairs of steam plant	£ 7000	£ 10500
Total per annum (steam)	£ 15800	£ 23700
Charges for extra cost of motorship	£ 15000	£ 15000
Advantage of motorship, per annum	£ 800	£ 8700

The comparative cost of motorships and steamships is certainly not well established by Mr. Johnson, who merely states that, "at the time the contract for the *Duchess of Bedford* was placed, prices were taken from leading shipbuilders for quadruple-screw Diesel installations * * * (which) showed that the cost of a motorship would have been £100,000 more." This is a large amount and deserves itemizing. What, for instance, is the entire cost of Diesel propelling engines? Unless separate ship plans were drawn, favorable design features effecting either a reduction in the size of the motorship or an increase in its passenger and freight capacity could hardly be evaluated. Also, a twin-screw installation would be less costly and readily available, today.

Professor Hillhouse, in his paper recently read before the Institution of Engineers and Shipbuilders in Scotland, showed the wide divergence in the comparative size of motorship and geared-turbine steamer, to meet the commercial demands of a given service:

	Oil Engines	Single-reduction geared turbines and cylindrical coal-fired boilers
Length of load waterline	480 feet	520 feet
Length between perpendiculars	460 feet	500 feet
Breadth	65 feet	70 feet
Depth	38 feet 3 inches	38 feet 5 inches
Draft	24 feet 8 inches	24 feet 8 inches
Block coefficient (on L.W.L.)	0.7175	0.725
Displacement	15,670 tons	18,475 tons
Weight of hull	5,370 tons	6,400 tons
Weight of machinery	1,500 tons	1,450 tons
Weight of refrigerating machinery and insulation	2,500 tons	2,500 tons
Weight of fuel	950 tons	2,650 tons
Weight of cargo, deadweight	5,000 tons	5,000 tons
Weight of crew, stores and water	350 tons	475 tons
Gross tonnage	10,250 tons	12,000 tons
Net tonnage	6,500 tons	7,700 tons
Brake or shaft horsepower	7,750	8,300
Length of engine room	70 feet	45 feet 4 inches
Length of boiler room	.....	58 feet 8 inches
Length of cross bunkers	.....	18 feet 8 inches
Total length of machinery space	70 feet	122 feet 8 inches

The conclusion drawn from his study was that the initial cost of steamer and motorship would be about the same. More especially is this true of passenger liners, the difference in machinery cost affecting less the larger cost of the ship. The reported opinion of Mr. Sterry B. Freeman, superintendent engineer of the Blue Funnel Line, that the cost of several sister ships—turbine and single-acting Diesel engine—was within 3 per cent of each other, should do much to dispel misapprehension concerning the cost of motorships.

Too often, after steamship plans have been drawn, alternate bids on Diesel propulsion of the steamship hull are casually invited from shipbuilders, who, apprised of the owner's preference by the steamship design, are naturally disinclined to incur heavy expense in preparing detailed estimates under such circumstances.

The basis for most of Mr. Johnson's comparisons is the record of the *Duchess of Bedford;* and it is not clear why in Table III he gives the propelling machinery a service rating of 16,000 shaft horsepower, whereas in his subsequent comparisons he figures it as 18,000 shaft horsepower. Clearly this affects such items as unit weight of the machinery, which he indicates to be 282 pounds per shaft horsepower on the 18,000 shaft horsepower rating; but which would be 317 pounds on the 16,000 shaft horsepower. Then, in Table XIII, he shows the weight of machinery in an equivalent motorship to be 410 pounds per shaft horsepower, which is far in excess of the figure for large motorships of modern design, for which 340 pounds per shaft horsepower would be more nearly correct; and it is safe to say that the weight of the Diesel installation will continue to be reduced, as also that its efficiency will continue to be increased.

The information in Mr. Johnson's paper concerning the precautions necessary to exclude air, scale-forming matter and oil from the boiler water is, undoubtedly, of great value to shipowners contemplating the building of ships fitted with high-pressure steam equipment. He fails to make com-

parison with the Diesel plant, which, in contrast, large or small, is extremely simple; and its present development is along lines of greater simplicity, lower weight and cost.

Mr. Johnson's assumptions as to the comparative fuel cost and first cost of motorships are so lacking in fairness to the new type of internal combustion marine power plant as to recall the adage, "There are none so blind as those who won't see." Shipowners will do well to seek more authentic sources of motorship information, in visioning the passenger and cargo liner of ten to twenty years hence, which sound economics will provide with internal combustion, requiring substantially less fuel than for external combustion under steam boilers, subject to further improvements and lowered costs of both types of plant, which will naturally become available as time goes on.—"Marine Engineering and Shipping Age," June, 1929.

ADVANTAGES OF HIGH-PRESSURE BOILERS.

By WALTER M. MCFARLAND* and T. B. STILLMAN.†

Mr. Johnson's paper on "The Propulsion of Ships by Modern Steam Machinery" is one of the best papers that we have read on this subject in some time. The author covers the subject very thoroughly and states the facts as he finds them, in a clear-cut way. Because of the excellence of the paper, there is very little of a critical nature which could be mentioned in discussing it, but there are a number of points which we believe should either be amplified or emphasized to bring out their true value.

We believe the story would be more complete, if, in mentioning the types of steam plant which the writer refers to as "prominently in use and applicable over a wide range of tonnage," he included electric drive and double-reduction gearing. The advantages of the former, especially in the larger sizes of passenger ships, is a very important factor in modern marine propulsion, as indicated by the results being secured on the S. S. *California* and the S. S. *Virginia*. The double-reduction gearing for turbine installations, especially for the moderate powers, has proved to be the most efficient type of steam propulsion which can be employed, and the reliability of some of the later installations, such as that on the S. S. *Dixie*, has proved that it may thoroughly be depended upon. Although we know of no large installations at the present time using double-reduction gearing, at the marine meeting of the American Society of Mechanical Engineers held on April 29 the point was clearly brought out that manufacturers of this type of equipment are now prepared to go to large powers and assure absolute dependability, together with the high economy inherent in this design.

Mr. Johnson's discussion of the condenser problem is excellent and very pertinent to the subject of the higher steam pressures and temperatures which are to be generated by the use of watertube boilers. One feature he did not mention, which is receiving consideration now in the marine field, and which has been used for years in stationary practice, is the division of the water space of the condenser into two units so that at any time, if a leak develops in a condenser tube, half of the condenser may have the water diverted from it (slowing down the ship a trifle due to the lack of higher vacuum), while the leak is being located and plugged. A construction such as this would make possible the ready location and plugging of a condenser tube in a ship at sea where only one condenser is in use, eliminating the

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necessity, so frequently followed at the present time, of running a leaking condenser until the ship reaches port. In this way any possibility of a badly salted-up system could readily be avoided.

It would also be desirable to have electric salinity indicators attached to the condenser discharge so that the operators could be instantly advised when a leak develops, in order that proper steps could be taken to shut off the side of the condenser affected and repair it before an appreciable amount of salt got into the feed-water system.

It is interesting to note that the use of Scotch boilers in place of the original inefficient watertube boilers of the S. S. *Empress of Australia* made possible the elimination of 1500 tons of dead ballast which had been carried in the ship with the watertube boilers. Fifteen hundred tons is quite an item of non-productive weight for a ship to be carrying around, and it is just as non-productive whether it is in the form of Scotch boilers or pig iron.

Although the steam pressure on this ship is moderate, the superheat is fairly high, being 230 degrees F., the latter undoubtedly being an important factor in securing the excellent performances, shown in Table No. 3, reported for this ship. It is also interesting to note that the auxiliary load on this ship is carried by Diesel engines, causing a very low auxiliary figure for oil per shaft horsepower, this being 0.06 pound, which is only 8.5 per cent of the total fuel burned. The use of these Diesel-driven auxiliaries produces a highly satisfactory oil per shaft horsepower figure for all purposes (0.71 pound), but it is to be questioned whether the method used to secure this low figure is justified in view of the expense of a Diesel installation as well as the cost of the Diesel oil which the auxiliaries require. It would undoubtedly be better, from a commercial point of view, to generate this auxiliary power from the main units, either from a generator geared to the main unit in case of a geared-turbine installation, or directly from the electricity generated by the main unit in the case of an electric-drive installation. These latter methods would also eliminate the complications inherent in having Diesel experts aboard the ships for auxiliary purposes only, and the necessity of a separate system of tanks devoted to carrying Diesel fuel and lubricating oil.

The comparison of the performances of the *Duchess of Bedford*, and the *Empress of Australia* shows the advantages to be expected from the use of higher pressures. The former, employing 340 pounds and 237 degrees F. superheat, used 7.3 pounds of steam per shaft horsepower for the main turbines, while the latter, with 220 pounds pressure and 230 degrees F. superheat, used 8.3 pounds of steam per shaft horsepower. The corresponding oil rates per shaft horsepower, for these two ships' main propulsion only, were 0.57 pound and 0.65 pound, respectively.

In addition to the saving in fuel oil indicated above, which amounted to approximately 12 per cent, there was also, as Mr. Johnson pointed out, a saving in weight of approximately 12 per cent and a saving in space of 16 per cent due to the use of the higher-pressure installation in the *Duchess of Bedford*.

Mr. Johnson very clearly brings out the advantages of watertube boilers as compared to Scotch boilers, showing that the installation costs of the high-pressure watertube type are less than a similar installation of the lower-pressure Scotch boilers. He also brings out an excellent point in showing the advantages of using large watertube boilers in place of a multiplicity of smaller units; reductions in weight, cost of boilers and cost of piping being the essential features of this comparison in favor of the larger units. With the modern methods of boiler care and operation at present being followed by up-to-date operators, there is no reason why the larger units should not be used and the corresponding benefits to the owners secured.

As a concrete illustration of the advantages to be expected from the use of larger boilers in a ship, it may be said that if six Babcock & Wilcox marine-type boilers are replaced by twelve of the same type, both installations having approximately the same total heating surface, the increase in weight would be 24.2 per cent, and the increase in cost approximately 18 per cent. The above figures are based on the use of boilers designed to operate at 400 pounds pressure and 250 degrees F. superheat. The heating surface per boiler of the smaller units is 3445 square feet, and of the larger units 6940 square feet.

Mr. Johnson has very clearly pointed out the advantages of keeping the water used in high-pressure and high-temperature boilers in excellent condition to save the costs and disadvantages of frequent boiler outage. He suggests, as one method of obtaining this result, running all of the make-up water through low-pressure Scotch boilers, the steam from which is employed in heating the ship and running auxiliaries which are steam driven. This is one way of doing it, but it seems like rather a backward step to put in high-pressure boilers and turbines for main propulsion, and then revert to Scotch boilers to insure satisfactory feed-water conditions. We believe that it would be more desirable under such conditions to follow the procedure of the modern central stations ashore, and run the make-up feed through high-pressure evaporators, using the saturated steam from these evaporators for heating and the auxiliaries that need it. This would save weight, and the evaporators would be very much more easily cleaned, when such a procedure is necessary, than the Scotch boilers.

Either system is to be preferred, where high-pressure and high-temperature boilers are in use, to the one generally followed of taking saturated steam directly from the main-boiler drums for auxiliary and heating purposes, because of the wide fluctuations of the superheat which frequently occur in service when the latter procedure is followed. In cases where high-pressure evaporators are not used, and water is obtained for boiler feed from low-pressure evaporators, it would then be desirable to use de-superheaters (preferably the coil of pipe in the boiler-drum type) for obtaining the saturated steam necessary for auxiliary purposes.

In Mr. Johnson's comparison of Diesel and steam propulsion, it is unfortunate that he did not present some actual Diesel upkeep costs, as well as those he gave for boiler upkeep, over a period of twenty-five years. This item of Diesel upkeep is one of the most important to be considered in any study involving the comparison of the two methods of propulsion, and although it is interesting to note that steam is a better commercial installation, assuming that there are no Diesel upkeep expenses whatever, the presentation would have been very much stronger, if the author had presented some exact figures of this very important item.

The direct comparisons Mr. Johnson gives, obtained from the operation of his *Beaver* class of ships, is most interesting and valuable as showing the advantages of mechanical stokers in place of hand firing. His results show a saving of approximately 10 per cent in coal due to the use of the stokers, and, as he indicated, this was accomplished with an appreciable reduction in the labor of the fireroom force. Although Mr. Johnson did not mention it, it is usually possible, in the average port, to secure the small-size coal desirable for stoker operation at a lower cost per ton than a corresponding grade of coal involving larger sizes such as the run-of-mine. A further advantage of the stoker installation is, as Mr. Johnson points out, the steady steam pressure and temperature conditions obtained with the stoker-fired boilers compared with that secured with similar hand-fired installations.

We feel that Mr. Johnson is quite correct in his closing statements to the effect that we may anticipate operating our ships in the near future at pressures from 500 to 600 pounds and temperatures up to 750 degrees F. This development will take place, with its corresponding economies in fuel consumption, just as soon as the operators have learned to handle their boiler feed-water conditions properly, as the men in the stationary power plant field have learned to do during the last few years.—"Marine Engineering and Shipping Age," June, 1929.

FACTORS AFFECTING CHOICE OF PROPELLING MACHINERY.

By JOHN F. METTEN.*

Mr. Johnson's paper should be carefully considered by all owners who are contemplating building new tonnage. It gives a logical discussion of the fundamental factors that should govern the selection of a ship's machinery installation. The treatment of the subject is unusually valuable in that the operating relationship between the various components is clearly illustrated, and the practical considerations that should influence general design are given proper weight.

The installation of the separate cylindrical boiler for taking care of the steam supply to oil-tank heating coils and other sources of feed water contamination solves one of the principal operating problems in connection with the adoption of higher pressures and temperatures of steam for the ship's power plant.

It should be particularly noted that the *Duchess* class of ships, of approximately 18,000 shaft horsepower, are showing an actual consumption for all purposes of less than 0.65 pound of oil per shaft horsepower and that still lower consumption figures may be expected in the later *Empress* class when they are put in service. It is probable that the new German transatlantic liners that are expected to go in service this year will also equal these figures.

Such results are being obtained by adopting the best proven marine practice and combining the knowledge and experience of the owner's operating force with that of the most experienced and successful shipbuilding yards. Practically all of the new post-war foreign tonnage has been built up to its present commanding position in this way. These owners have tried out all of the various innovations in ship's propulsion machinery that are being systematically advocated in the United States for its new merchant construction, and have apparently discarded features that in any way fail to qualify in the three marine engineering graces of reliability, operating economy and minimum first cost—and the greatest of these is reliability.—"Marine Engineering and Shipping Age," June, 1929.

GEARED VERSUS ELECTRIC DRIVE FOR MERCHANT VESSELS.

By A. PETERSON, DE LAVAL STEAM TURBINE CO.

The choice of propelling machinery for merchant ships is just now widely discussed by marine engineers and others, having reached even the columns of the "Saturday Evening Post." This discussion began with the introduction of the steam turbine and the internal combustion motor into marine service and each new development has opened up further possibilities for

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difference of opinion. The reciprocating steam engine remains about as it was, but internal combustion advocates now have two cycle and four cycle systems, directly connected, with electric transmission and geared, to choose from, while the steam turbine camp, which has long since abandoned directly-connected turbines, disputes the relative merits of mechanical speed reducing gears and electric transmission.

The arguments advanced fall into two classes, empirical and theoretical. The former quote overall results actually attained in the way of first cost, reliability, absence of noise and vibration, fuel consumption, weight and space requirements, upkeep and attendance, etc., arguments which appeal to owners and operators, particularly to non-technical men who are not equipped, or inclined, to analyze the operation of power plants and to determine what is essential and peculiar to a given system, as against what may be immaterial or generally applicable to all systems.

The theoretical method of argument, which is suitable for engineers studying engineering problems, is to divide out the non-essentials from the essentials and to draw exact comparisons between parallel component parts or assemblies of the several systems, making suitable allowances where changes in one part of the equipment affect other parts. For example, the boilers and auxiliary equipment with a geared turbine installation develop less power and require less fuel than is required for an electric drive installation of the same power. Roughly speaking, about half of the advantage of geared drive with respect to first cost and fuel consumption comes from the requirement of less auxiliary power. Such substantial advantages should not be ignored.

The only accurate way to make such a comparison is to compare complete machinery installations, taking everything into account, and including differences in propeller revolutions as well as in auxiliaries, where such exist, as for example, on vessels of slower speeds.

In an article entitled, "Steam Turbine Electric Drive for Merchant Vessels," by H. C. Berrian, Assistant Chief Engineer, Newport News Shipbuilding & Dry Dock Co., which originally appeared in the May, 1929, issue of the JOURNAL OF THE AMERICAN SOCIETY OF NAVAL ENGINEERS, and which was reprinted in the July 1st issue of "Marine Journal," arguments based on both overall results and analysis of performance, as well as some analogies between central stations and marine power plants, are presented in such a way that geared turbine drive for merchant vessels appears to be discredited, while turbine electric propulsion is promoted.

The overall results arguments adduced by Mr. Berrian can be offset by numerous instances which favor gear drive. For example, he mentions the introduction of electric propulsion in the United States Navy as a subject which presumably needs no further discussion, but does not mention the fact that there are now in commission in the United States Navy ten 7500-ton light cruisers having four shafts and normally rated at 90,000 shaft horsepower each, all propelled by geared turbines, nor that there are now building eight 106,000-shaft horsepower cruisers, while Congress has authorized 15 more of the same type, as well as a 60,000-shaft horsepower twin screw, air craft carrier, a total of 24 new ships to be propelled by geared turbines, nor that the British, French, Italian and Japanese navies have many high powered naval vessels of all types propelled by geared turbines, a large number being of over 100,000 shaft horsepower. Mr. Berrian himself undoubtedly would not cast any doubt on the efficiency or reliability of the De Laval gears installed by his own company in the cruiser now building at Newport News, nor would he imply that the successful cutting of such 26,500 horsepower gears, as accepted by the shipyard and by the Navy, was at all problematical.

All together, the United States Navy has over 10,000,000 shaft horsepower of geared turbines, as against a total of only 632,000 shaft horsepower of turbine electric drive in navy, cost guard and merchant marine, together, over half of which is in two ships, neither of which, according to current reports, stands strictly at the top in respect to either economy or reliability. Four reputable gear manufacturers in the United States, namely, De Laval, Falk, General Electric and Westinghouse, have millions of horsepower of successful marine gear installations in operation in practically every type of seagoing vessel, including almost 300 installations of 28,000-shaft horsepower gears in twin screw, turbine driven destroyers in the United States Navy. The foregoing facts leave little force in Mr. Berrian's remark that "as the power of the geared steam turbine increases, it gradually becomes increasingly difficult to provide adequate and reliable reduction gears."

It is not sufficient to claim that "the principles involved in electric transmission on shipboard are identical with those which have been thoroughly worked out in central station practice," and that "it is quite natural that the United States, which has been the leader in electrical development on land, should be the leader in electrical development on shipboard," since many circumstances of power application on shipboard differ from corresponding land practice. Land central stations are nearly always interconnected, and the turbine generators are invariably run at constant speed, 60 cycles frequency being most generally used, which sets the turbine speed at 3600, 1800 or 1200 revolutions per minute. Central station turbine generators are, further, frequently designed, in order to make the most favorable combination of efficiencies, speeds, loads, weights and costs, to pass through one, two or even three critical speeds in reaching the standard running speed. Again, the load absorbing appliances on land are as a rule comparatively small and scattered over a wide area, whereas the marine load absorber is located near the turbine, and takes the entire output. To employ all the complicated appliances required in the distribution of electric power over hundreds of miles to many small consumers for the purpose of transmitting a single large unit of power over a few feet does not seem "natural" when there is at hand so simple and reliable a means of transmission as the mechanical speed reducing gear.

It is generally accepted as good practice in designing marine propelling machinery to have no critical speed within the maximum propeller speed of the unit. The marine prime mover, except for direct current turbine generators, which are restricted to quite low powers, must necessarily be capable of continuous operation at variable speeds. Contrary to an apparently widely held belief that electric drive turbines always run at constant speed even when the propeller runs at reduced speed, the fact is that the ratio of turbine speed to propeller speed is fixed by the respective numbers of poles in generator and motor. In order to keep down the weight, cost and size of propelling motors so that they can be used to propel ships, the minimum possible number of poles in the motor is desirable. The mechanically geared turbine does not suffer from any such limitations of speed and ratios. By the use of compound or triple expansion turbines, the turbine shafts can be made short enough so that the critical speed will be above the maximum running speed, no matter how high the turbine speed nor how low the propeller speed may be set for reasons of efficiency. The speeds of electric drive turbines appear on the other hand to have been selected in accordance with land designs.

Mr. Berrian's statements that the designer of turbine-electric propulsion equipment finds himself with a free hand in determining steam conditions and that the single, relatively high speed, turbine rotor constantly revolving

in one direction has been thoroughly worked out in land practice for pressures and temperatures in excess of anything that he will probably care to adopt, carries the implication that the geared turbine is somehow or other inferior in these respects. The fact is, however, that the turbine is not the limiting factor with respect to temperature, the boiler, superheater and piping setting the limits in this respect. Furthermore, the following geared turbine vessels have much higher steam conditions than do any electric turbine equipments now in service:

S. S. *Dixie*, 350 pounds per square inch, 200 degrees F. superheat, 28½ inches vacuum.

S. S. *King George V*, 550 pounds per square inch, 800 degrees F total temperature, 28½ inches vacuum.

Duchess class ships, 350 pounds per square inch, 250 degrees F. superheat, 28½ inches vacuum.

Experienced geared turbine manufacturers are prepared to build marine units to run satisfactorily at any steam pressure or temperature which reputable steam boiler manufacturers are willing to supply.

Regenerative heating of feed water by means of steam bled from the main turbine is a feature common to both systems of turbine drive, and the same methods of generation and distribution of power to auxiliaries may be used in either case. There is no question about the desirability of practically complete electrification of deck and engine auxiliaries, and nearly all reputable marine engineers recognize their efficiency, simplicity and reliability. Electric auxiliaries supplied with current from small, high efficiency condensing turbine generators are just as efficient on geared turbine vessels as on turbine electric vessels, and in addition, the geared turbine drive does not require current for excitation as does the turbine electric drive. In the case of the S. S. *California*, for example, additional auxiliary power amounts to 185 kilowatts for excitation and ventilation, corresponding to about 2900 pounds of steam per hour.

The suggestion of dual drive of the auxiliary generator by an auxiliary turbine and by a synchronous motor receiving current from the main units on the turbine electric vessel is paralleled on the geared turbine ship by direct coupling of the auxiliary generator to the main unit, with the avoidance of generator and motor transmission losses.

As regards space occupied by propelling machinery, the engine room length on the *Virginia* is stated to be about 6 feet less than would have been required for a geared turbine installation. The gear drive installation compared with the *Virginia* was probably not well designed, because in nearly all cases gear drive machinery requires no more, and usually less, space.

In respect to weight, the geared turbine with double reduction gears shows up still better, assuming the proper turbine speeds for maximum economy and reliability with low propeller speeds. Where the propeller speeds necessitate only one step of gearing, the geared turbine generally weighs only one-half as much, or even less, as does the turbine electric drive for the same propeller speeds and steam conditions, including in the comparison turbines, generators, motors, lubricating system, exciter and foundations, cables, main control board, Kingsbury propeller thrust bearings, proportionate amount of additional weight for auxiliary condenser due to excitation and proportionate extra weights of main boilers and condensers required by the turbine electric drive ship, due to lesser inherent efficiency of electric transmission.

As an example of the weight and space requirements of the two forms of drive, should the Navy decide to put geared turbine propelling machinery in the *Saratoga*, reliable, efficient geared turbines with condensers and con-

denser auxiliaries could be placed in the space now occupied by the main motors alone, and would weigh substantially the same as these motors, while the weight represented by the present generators, cables, controls, exciters, ventilating blowers, foundations, condensers, etc., including two main boilers, would be saved.

The claim that there is no appreciable difference in fuel consumption as between geared turbine and turbo-electric propelling machinery of the same power and for the same steam conditions, propeller revolutions and safety factors ignores the fact that transmission through speed reducing gears is 3 to 5 per cent more efficient than through generator and motor, and even more when the excitation for generator and motor and ventilation for the motor are included.

In comparing turbine with turbine, it is necessary to make certain assumptions in reference to the type of turbine that is to be used in each case. The main turbine for ship drive should be built with a stiff shaft, as flexible shaft turbines are not rugged enough to be satisfactory on board ship. It is not possible to build a single shaft turbine of 8000 horsepower or larger with a stiff shaft that will give as high economy as can be obtained in a two or three casing turbine, such as would be used as part of a geared turbine propelling unit. The advantages gained by the multiple casing turbine through its ability to use different and suitable speeds for the high and low pressure parts and to use more stages, while still retaining the stiff shaft feature, more than compensate for losses due to additional stuffing boxes, to a constantly rotating reversing element and to steam friction through cross-over connections.

If the proper practice in design were followed, all electric drive turbines would be of the compound type in order to secure a flat fuel rate curve. It is impossible to obtain a flat fuel rate curve with a single turbine and with the electric drive turbines as at present designed. For both high pressure and low pressure turbine elements to run at the same speed is disadvantageous for the high pressure turbine. On the other hand, geared turbines are designed for exactly the best speeds and the high pressure turbine runs at a higher speed than the low pressure turbine.

The superiority of the geared turbine with respect to fuel consumption is well borne out by the attested performances of ships now in regular service. Mr. Berrian cites the fuel consumption for all purposes of the *California* as .764 pound per shaft horsepower, and that of the *Virginia* and *Pennsylvania* as .723 pound, the *California* using steam at 275 pounds pressure and 100 degrees F. superheat, while the *Virginia* and *Pennsylvania* are designed for 300 pounds pressure and 200 degrees F. superheat. As against these figures we have the performance of the *Dixie*, of about half the tonnage and power, of .783 pound of oil for all purposes with steam auxiliaries, which would become .66 pound of oil per shaft horsepower for all purposes if the *Dixie* were equipped with electric auxiliaries of the same type and efficiency as those used on the *California*.

Mr. Berrian further estimates that if the *Virginia* and *Pennsylvania* were equipped with air heaters and high pressure bleed feed water heaters, the oil consumption could be brought down to .65 pound per shaft horsepower. In comparison with this figure may be placed the estimate of a reputable ship builder on double geared turbines for a recent twin screw job requiring approximately 22,000 shaft horsepower of an oil consumption of .55 pound per shaft horsepower for all purposes, based on steam at 375 pounds gauge and 725 degrees F. total temperature, with 28½ inches vacuum. The builder further offered to guarantee this estimate with a margin of 6 per cent.

The claim of superior reliability for the turbine electric drive is borne out neither by actual performances of ships in service nor by arguments based

upon the combinations which can be made between two turbine generators and two or more electric motors. In a single screw ship, where one generating unit is provided, the compound geared turbine has some advantage, as it can be separated into two turbines, either of which may be used independently of the other. Compound geared turbines are provided with blind flanges, and the piping is so arranged that either the high or low pressure turbine can be cut off and the other operated at half power. In the twin screw ship, where two generating units are provided to develop full power, turbine electric drive has certainly no advantage over two compound geared turbines.

In general, reliability depends more on conservative design and ample factors of safety than on anything else. Presumably for the purposes of reducing the price differential to a minimum, while striving for good steam economy, the turbines offered for marine electric drive have been designed with factors of safety and shaft characteristics such as are used in constant speed land practice.

Marine service is unquestionably much more severe than land service and requires a much greater margin of safety if wheel rupture, loss of turbine buckets, etc., as reported from several notable recent electric drive installations, are to be eliminated.

Geared turbine machinery is more simple in operation than is the turbine electric drive, since the latter contains many more elements requiring supervision, adjustment and overhaul. As for the alleged advantages of electric meters, these are subject to errors such that they cannot be relied upon in detecting small changes in efficiency of the units. When acceptance tests are run on central station equipment, the station instruments frequently do not agree with the manufacturer's instruments and often neither set agress with specially calibrated instruments obtained from the testing laboratory. It may be noted, also, that in a recent paper, R. V. Smith gives a different propulsion efficiency for the *California* than is given in the paper on the same vessel by Capt. Roger Williams. As for reading off power being developed at any instant, this is, with a given turbine, quite definitely determined by the steam conditions and speed, and may readily be tabulated or shown in curves for the convenience of the men in the turbine room.

The comfort of passengers and crew is much talked about by electric drive advocates, but there is as much freedom from vibration with present day geared turbines as built by manufacturers who have proper facilities and experience as with turbine generators and motors built by anyone. The windage noise of turbine generators and motors is as great or greater than the noise of properly cut gears as manufactured by experienced concerns. This fact may be demonstrated by a visit to the S. S. *Dixie* or the S. S. *Waialeale*, the geared turbines of which cannot be heard outside of the engine rooms. The possibility of propeller racing is no different in a geared ship than in one in which the propeller is coupled to the generator through a synchronous motor, and in either case can be controlled by a speed governor.

The condenser arrangements described by Mr. Berrian are not in any sense peculiar to the turbine electric generator, and were first used in marine practice in connection with geared turbines, as, for example, in the Japanese, British and Chilean destroyers and cruisers, on the *Duchess* and *Empress* classes of ships of the Canadian-Pacific S. S. Co., in the steamships *Dixie* and *Waialeale*, etc., and altogether there are some 50 geared steam turbines in American vessels having bottom exhaust and underneath condensers.

This question of marine propulsion should be studied by experienced, unbiased engineers, such as the technical staff of the U. S. Shipping Board,

and recommendations made based upon all the facts, rather than upon the "appeal of the all electric ship," coupled with highly efficient publicity efforts.

We hear continually that the American Merchant Marine is handicapped because American ships cost more to build. Geared turbine machinery is superior for merchant vessels because, everything considered, it will earn higher dividends. This results from the following:

1. The first cost of the machinery installation is from 5 to 15 per cent less.
2. The fuel consumption is considerably less.
3. The reliability of the geared turbine is greater for the reason that the factors of safety are much higher than those of the electric drive turbines and there is less complication.—"Marine Journal," July 15, 1929.

THE LENTZ STANDARD MARINE ENGINE.*

By Mr. W. J. Muller.

In 1924 the board of directors of the Koninklyke Paketvaart Maatschappy, of Amsterdam, decided to enlarge their fleet plying in the Dutch East-Indian Archipelago with a large number of vessels of different size and speed. It was obvious that, on doing so, there would be an advantage in dividing the vessels into certain classes of uniform ships and therefore to standardize the ships and the machinery equipment as far as possible. Besides a number of other ships with different types of engines, twenty-three steamships, one twin-screw passenger steamer included, with twenty-four steam engines in all, were to be built, according to the program of 1924, in the course of five years.

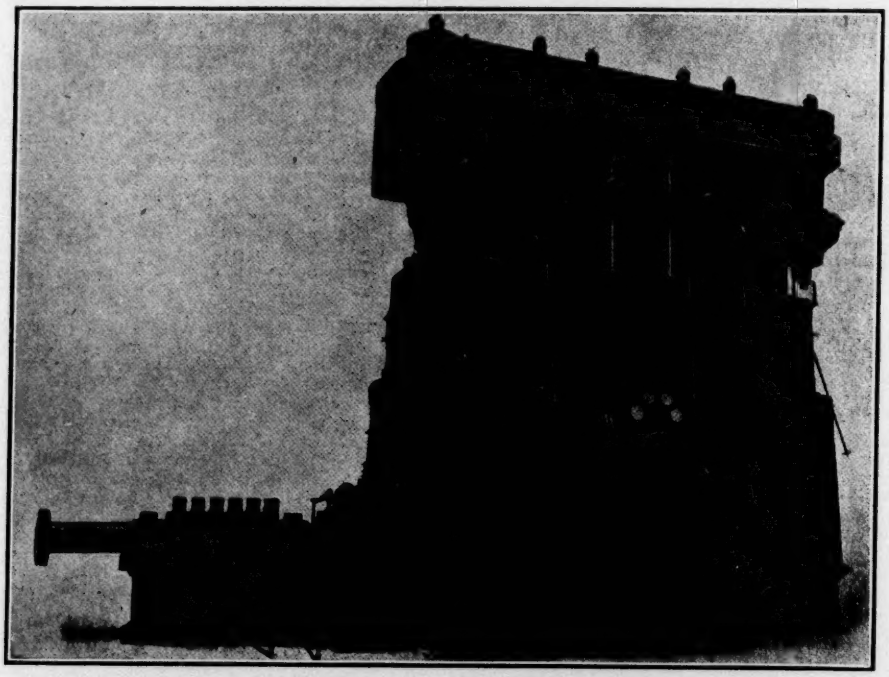
In the beginning of 1924 we had an opportunity personally to make a trip in the steamer *Bilbao* of the Oldenburg-Portuguese Shipping Company, of Hamburg, which vessel must be regarded as being the first ship equipped with a Lentz standard marine engine. As a result of this trip, the performances of the main engine, and the good results as regards coal consumption and costs for upkeep and repairs, we closely considered the application of the Lentz standard marine engine on a number of K.P.M. vessels.

It was decided in 1924 that five ships, to be followed later on by eighteen ships, were to be installed with Lentz standard marine engines. For this purpose the K.P.M. bought the patent rights for the application of this type of engine on the company's own ships from Messrs. W. Salge and Co., Berlin, owners, of the Lentz patent rights as far as the Lentz standard marine engine is concerned, in order to be free to have the engines built by any engine maker in Holland. The actual engines were built by Stork Bros., of Hengelo, and Werkspoor, of Amsterdam.

GENERAL DESIGN.

The Lentz standard marine engine has been made in six standard sizes, ranging from 400 I.H.P. to 3700 I.H.P. Assuming a boiler pressure of 206.2 pounds per square inch and a steam temperature of 617 degrees F. with a 90 to 92 per cent vacuum, the estimated steam consumption works out at from 9.26 to 9.74 pounds per I.H.P. hour for the smallest size of engine, and 9.02 to 9.48 for the largest size, the admission of steam in the H.P. cylinder being from 30 to 40 per cent. The ratio of the H.P. and L.P. cylinder volumes is about 4 : 6, and the total expansion of the steam in both the cylinders varies from 15.3 to 11—neglecting the clearance volumes. The

* Institute of Marine Engineers. Abridged.



1200-I.H.P. Lentz Standard Marine Steam Engine.
Stork Brothers, Hengeld, Holland, Engineers.

Lentz standard marine engine consists of two entirely separated halves, each forming a complete and self-contained two-cylinder two-crank compound steam engine, which halves are interconnected only by the crank shaft. The two H.P. cylinders are placed in the center, one L.P. cylinder is placed forward and one aft. The steam after having passed the main stop valve is divided into four flows, two for the forward engine-half (one for top-side and one for bottom-side of the cylinders) and two in a similar way for the aft engine-half. The steam passages are formed by ducts, which are placed so that the steam always passes in the same direction.

For steam distribution there are altogether twelve independent valves, *i.e.*, three for the top-side and three for the bottom-side, for each engine-half: the H.P. inlet valves A; the H.P. exhaust; also L.P. inlet valves B; the L.P.

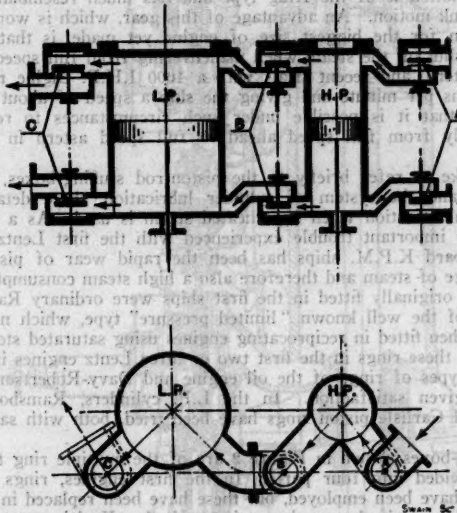


FIG. 1.—ARRANGEMENT OF VALVES.

exhaust valves C. These valves are placed in one line and are operated by a separate cam shaft for each engine-half—see drawing Figure 1. The valves A and B are of the same size for reasons of standardization; but, as will be shown later, this has been a mistake. There is no receiver, and valve B acts merely as an overflow valve from the H.P. cylinder into the L.P. cylinder, which is possible as the two cranks belonging to the same engine-half are placed at 180 degrees, so that the exhaust of the H.P. cylinder, top-side, may coincide with the admission of steam to the L.P. cylinder, top-side, and similarly for the bottom-side.

The cranks of each engine-half are placed opposite each other, whereas the cranks of one engine-half are placed at right angles with regard to the other engine-half, so as to afford regular working and certainty of maneu-

STORK BROTHERS, HENGELO, HOLLAND, ENGINEERS.

vering, but as a consequence the system is not balanced, either statically or dynamically. It is, however, possible to fit counterweights to the crank webs, which has been done in the largest size only.

Drawing Figure 2 shows a section of the H.P. cylinder with valve chests and valves, and also the position of the cam shaft and cams. The valves are placed in valve cages, fitted in the casting with two seats and kept to its place by the valve chest covers and by bolts, to ensure perfect tightening. They are of the balanced double-beat type and are held down to the seatings by means of separate spindles provided with springs. The valve spindles are actuated by separate tappets with rollers, also called compensators, and by this arrangement correct alignment of the valve mechanism is ensured. The compensators also serve to eliminate faults in the adjusting of the valve gear, caused by the rise of temperature of the engine when running.

The link motion is of the Klug type and has much resemblance to the Hackworth link motion. An advantage of this gear, which is worked easily by hand, even for the biggest size of engine yet made, is that it is not necessary to shut off the steam when maneuvering from full speed ahead to full speed astern, and recent trials with a 1600 I.H.P. engine running at 100 revolutions per minute and giving the ship a speed of about 10 knots, have proved that it is possible under such circumstances to reverse the engine actually from full speed ahead to full speed astern in about six seconds.

I should like to refer briefly to the piston-rod stuffing-boxes, the H.P. piston rings and the system of cylinder lubrication, three details which deserve special attention when superheated steam is used. As a matter of fact, the only important trouble experienced with the first Lentz standard engines on board K.P.M. ships has been the rapid wear of piston rings, causing leakage of steam and therefore also a high steam consumption. The type of rings originally fitted in the first ships were ordinary Ramsbottom piston rings of the well known "limited pressure" type, which never gave any trouble when fitted in reciprocating engines using saturated steam. The rapid wear of these rings in the first two or three Lentz engines induced us to try other types of rings of the oil engine and Davy-Robertson patterns which have given satisfaction. In the L.P. cylinders, Ramsbottom and Lockwood and Carlisle piston rings have been tried, both with satisfactory results.

The stuffing-boxes shown in Figure 2 are of the multiple ring type, each ring being divided into four parts. In the first engines, rings made of nickel-bronze have been employed, but these have been replaced in the later engines by close-grained cast iron rings of the Kreisinger type, with tangential cuts, allowing all the four parts to remain tightly fitted to the piston-rod, even when wear has occurred.

As regards the lubrication of the cylinders, two systems have been tried, and at this moment it cannot be said which of the two is superior. The first one tried is the ordinary system with an atomizer in the live steam line to the cylinders. It would also seem more sensible to lubricate a surface rather than a volume! The other system of lubrication, by means of small holes in the cylinder wall through which the oil is brought into the cylinder, has also been borrowed from the Diesel engine, and a number of Lentz engines of the K.P.M. have been fitted with this system of lubrication, the H.P. cylinders being lubricated at three points at half-stroke, and, in a similar way, the L.P. cylinders at four points. We have reasons to believe that with the latter system less oil is required for proper lubrication than with the atomizer system, and we have found that from 1.1 pounds-1.7 pounds per 1000 I.H.P. is sufficient when using the last mentioned system, whereas with atomizers 1.7 pounds-2.2 pounds is required.

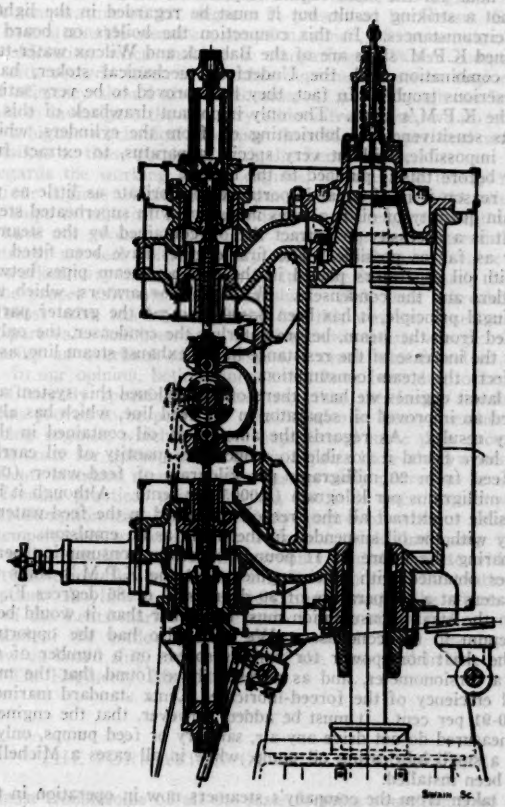


FIG. 2.—SECTION THROUGH H.P. CYLINDER.

OPERATING CONDITIONS AND STEAM CONSUMPTIONS.

The steam consumption, which has been measured on several occasions in engines of different sizes, we have found never to be under 11 pounds per I.H.P. per hour for the main engine only, without any auxiliaries, which is certainly not a striking result, but it must be regarded in the light of the operating circumstances. In this connection the boilers on board all the Lentz engined K.P.M. ships are of the Babcock and Wilcox water-tube type which, in combination with the Underfeed mechanical stoker, has never given any serious trouble. In fact, they have proved to be very satisfactory on board the K.P.M.'s ships. The only important drawback of this type of boiler is its sensitiveness to lubricating oil from the cylinders, which it is practically impossible, without very special apparatus, to extract from the feed-water before this is pumped to the boilers.

For this reason it is of great importance to lubricate as little as possible, but a certain quantity of oil is always necessary with superheated steam, and therefore it is a necessity to extract the oil contained by the steam or the feed-water as far as possible. The first engines have been fitted for this purpose with oil separators placed in the exhaust steam pipes between the L.P. cylinders and the condensers. With these separators, which work on the centrifugal principle, it has been possible to get the greater part of the oil separated from the steam, before entering the condenser, the only drawback being the increase of the resistance in the exhaust steam line, as this, of course, affects the steam consumption.

For the latest engines we have, therefore, abandoned this system and have now applied an improved oil separator in the feed line, which has also given satisfactory results. As regards the amount of oil contained in the feed-water, we have found it possible to reduce the quantity of oil carried over with the feed from 20 milligrams per kilogram of feed-water (0.002 per cent) to 5 milligrams per kilogram (0.0005 per cent). Although it has been found possible to extract all the free oil contained in the feed-water, this is not so easy with the oil suspended in the liquid as an emulsion.

In comparing the figure of 11 pounds of steam consumption per I.H.P. with figures obtained with other engines, that the K.P.M.'s ships trade in tropical waters at a temperature of 30 degrees C. or 86 degrees F., and for this reason the steam consumption must be higher than it would be with a higher vacuum in the condenser. We have also had the opportunity to measure the shaft horsepower for various speeds on a number of ships by means of a torsionometer, and as a result have found that the maximum mechanical efficiency of the forced-lubricated Lentz standard marine engine is about 90-91 per cent. It must be added, however, that the engines which we have measured do not drive any air, sanitary or feed pumps, only a bilge pump and a small lubricating oil pump, while in all cases a Michell thrust-block had been installed.

Records taken from the company's steamers now in operation in the East Indian trade—particulars of which are given in the paper—show steam consumptions for all purposes of 14.67 pounds for the smaller sizes of engine to 13.69 pounds per I.H.P. hour for the largest sizes. The consumption for the main engine also works out at 11.03 pounds.

The actual steam consumption, as we have found it, is higher than the consumption expected by the designers, Messrs. Salge & Co. Even when the resistance in the L.P. exhaust has been improved, we do not expect to find a better steam consumption than 10.8 pounds per I.H.P. for engines of the original design. We believe that this result can be improved by enlarging the size of the overflow valve and thereby diminishing the resistance in the steam passage between H.P. and L.P. cylinders, the drawback being, how-

ever, if doing so, that the clearance volume of the L.P. cylinder which is already on the high side, might be increased. Also the resistance in the L.P. exhaust valve might perhaps be decreased, as the velocity of the steam appears to be rather high, as in the H.P. exhaust valve. Also the maximum temperature of the steam on board K.P.M. vessels, 572 degrees F., is lower than the temperature of 617 degrees F. prescribed by Messrs. Salge & Co. for the best attainable steam consumption, which, of course, should be taken into consideration when making a comparison. When comparing these figures with records taken from other superheated steam engines, the mechanical efficiency should also be considered, and although we have not yet been able to investigate the ordinary slide valve engine, we do not expect it to have a higher mechanical efficiency than 87 or 88 per cent under similar circumstances to those obtaining with the Lentz engine.

As regards the working of the engines, we are able to say that all the twenty-four engines which we have put in service since 1926, when the first K.P.M. ships fitted with Lentz standard marine engines went to the East Indies, have not given any particular trouble up to the present time. On the contrary, we feel obliged to say that the valves and valve gear have always worked very satisfactorily; in fact, we know of only two cases in which the poppet valves have been involved. Only once a breakdown of an L.P. exhaust valve has been reported, but this was not a serious case, and after a spare valve had been fitted the ship continued its voyage. The second time, during an overhaul of the engine, a crack was found in one of the valves. In our opinion, both failures must have been due to faults of material. Generally speaking, we may say that the Lentz standard marine engine has proved to be perfectly reliable, also in heavy weather conditions with racing engines, which we have experienced more than once. Furthermore, the costs for upkeep and ordinary repairs have always been very low.

The Lentz standard marine engine has certainly proved to be a very good engine from a practical point of view, but it is questionable whether the steam consumption can be considerably improved without increasing the steam temperatures as well. Also I do not think that many large Lentz engines will be made, as the engine certainly cannot compete with a modern geared turbine installation of 3000 S.H.P. or more in the matter of steam consumption. Anyhow it will be interesting to watch the efforts of the designers of the Lentz marine engine and of those of the ordinary slide valve engine and of the uniflow marine engine, which latter type has also been tried now on board ship, to see which of the three will in the long run succeed.—"The Engineer," May 17, 1929.

THE FORMATION OF SCALE IN BOILERS.

The most troublesome of all the scale-forming constituents of feed-water is undoubtedly calcium sulphate. Sulphate scale is a hard, dense deposit almost like porcelain; it is particularly difficult to remove from the metal surfaces and is probably responsible for more bent, blistered or burst tubes than all other causes put together. The presence of calcium sulphate also renders other deposits more dangerous, for those that would be porous or friable by themselves are entangled by its needle-like crystals and consolidated into a hard layer capable of giving more trouble than either deposit singly. The importance of calcium sulphate as an impurity of feed-water has led to numerous studies of its nature and effects. Three forms of it are found in boiler-scale, ranging from gypsum with two molecules of water

of crystallization to the anhydrous salt with none. All three forms produce deposits which are essentially similar. The first two, however, are associated with temperatures lower than those met with in modern steam practice, and as they both tend with time to pass automatically into the anhydrite, the latter alone is of practical importance to boiler users. All forms of calcium sulphate are thrown rapidly out of solution as the temperature rises, water which can hold about 650 parts of the anhydrite per million at atmospheric boiling temperature depositing as much as 91.5 per cent of the salt by the time a temperature of 220 degrees C., or a boiler pressure of about 320 pounds per square inch is reached.

Two theories have been advanced as to the way in which the sulphate is deposited. According to one belief, as soon as the water reaches its limit of saturation, the salt comes out of solution in the form of minute particles distributed through the mass of the water, which eventually attach themselves to the heating surfaces and build up a thickness of scale. This, which may be called the colloid theory, has given rise to various attempts to prevent scale either by maintaining an electric potential tending to oppose the contact of the charged particles with the metal, or by adding other colloidal solutions with the idea of preventing the sulphate particles sticking together or to the metal surfaces. The alternative theory, which is that the scale is formed in position, appears to have been definitely established by Dr. E. P. Partridge and Professor A. H. White, who have embodied the results of some very interesting experiments in a paper contributed to the monthly journal of the A.S.M.E. By examining microscopically the higher polished surface of a metal plate through which heat was being transmitted to water containing calcium sulphate in solution, it became evident that scale always started as a fine ring of minute crystals, marking the place where a bubble had been detached. From these rings needle-crystals grew outwards into the solution, became interlaced with similar crystals from adjacent rings, and so on until the whole surface was covered with a crystalline deposit. Experiments with air and other gases in solution, which were liberated as bubbles before the boiling point was reached, confirmed the view that a bubble of any kind was sufficient to start deposition, though in boiler practice steam bubbles alone need be taken into account. According to the views of the authors the mechanism of scale formation in steam boilers is somewhat as follows:—When boiling temperature is reached the film of water next to the heated metal begins to vaporize at some point. The surface under the vapor then becomes hotter still, owing to the insulating effect of the vapor. This local overheating causes the film of solution surrounding the dry spot to be evaporated, the bubble therefore growing until it becomes detached from the metal by its own buoyancy. When such a process takes place in a saturated solution of calcium sulphate, the fluid film at the edge of the dry spot becomes supersaturated by the increase of temperature, and is forced to deposit minute crystals in the ring where metal, vapor and water meet. These crystals are always in contact with a supersaturated solution, first on account of the evaporation of pure water into the bubble; and secondly, by the return of liquid now supersaturated, after the departure of the bubble. They thus tend to grow continuously as long as heat enters the solution through the metal.

If this view is correct—and the experiments of Messrs. Partridge and White seem to leave little room for doubt about it—the colloidal theory is untenable, and electrical methods of preventing scale are founded on false premises. Many processes work, however, in spite of their theory being wrong, and investigations into electrical methods are being proceeded with. So far the laboratory results have been negative, the systems tried having no effect in stopping or decreasing the rate of deposition of scale. With

regard to the use of colloidal solutions as scale preventers matters seem more hopeful. The authors found that several of them were absorbed on the calcium sulphate crystals after the latter were formed on the metal, and strongly modified the physical characteristics of the scale. In some cases, indeed, the scale became so weak as to flake off the metal surfaces under the action of convection currents. It appeared probable, therefore, that some colloids might be beneficial in minimizing scale troubles, even though the theory advanced in support of their use was inaccurate. Another point also seems to have been cleared up, namely, the effect of the nature of the metal surfaces in respect of scale deposition. Suggestions have been made that scale would not form on evaporative surfaces of particular metals. Messrs. Partridge and White have, however, produced exactly similar scales under similar conditions upon surfaces of polished steel, oxidized steel, chromium, silver, nickel, and glass. They also discovered incidentally that in no case did calcium sulphate by itself produce a scale which appeared to be bonded to any degree to the surface on which it had formed. The scales invariably cracked off cleanly. The adherence of calcium sulphate scale, which is so strongly marked in practice, is thought to be a bonding effect due to the roughness of the tube surface and the mechanical strength of the scale.

If we are to accept the authors' theory that scale only starts at points where bubbles are formed, there follows an interesting consequence which has escaped their attention. When steam is generated by heating water at a pressure above the critical, as in the Benson boiler, no bubbles are or can be formed. Hence, if the theory is correct, no scale could be formed in a Benson boiler. Any calcium sulphate present would, however, have to be thrown out of solution at the high temperature of the water, so that it must either pass out in the stream as dust or remain in the boiler in the same form. There are few boilers working above the critical pressure in existence, and in such as there are it is probable that nothing but distilled water is used as feed, so that experience is wanting as to whether scale can be formed in them or not. If it can be the bubble theory would seem at least incomplete, while if it can not, then the type of boiler in question has at least one inestimable advantage over ordinary steam generators. The difference between the point of view of a laboratory and a boiler-house is shown rather strikingly by the discussion by the authors of the rate of formation of scale. They reason that as the quantity of calcium sulphate which water can hold diminishes rapidly with the temperature of the water and therefore with the pressure of the steam, a boiler working at high pressure will make less scale than if the pressure were lower. In support of this contention they quote experimental figures showing that with a constant rate of heat transference a boiler which deposits 0.320 pound of scale per square foot per day at 10 pounds pressure, deposits scale at only one-tenth of this rate at 50 pounds pressure, and at only 0.010 pound per square foot per day at 150 pounds pressure. This may all be perfectly correct, for it is certainly true that when a boiler is evaporating water which can only hold a small quantity of sulphate in solution, the scale deposited per pound of water evaporated will be less than if the water held more sulphate. Engineers, however, look at the question in a different and more practical way. To them the sulphate contents of the feed-water is the only criterion of sulphate scale-formation, for it is clear that all the sulphate entering the boiler must be left there, entirely regardless of the pressure at which the boiler may be working. If the pressure is low the density of the water will be greater, but once the saturation limit is reached at any pressure, the sulphate will begin to come down as scale and will continue to do so at exactly the rate at which it is renewed by the feed.

The experiments quoted would seem to have been made under conditions in which the boiler was always fed with water at the steam temperature, the feed therefore carrying much less sulphate per pound of water at the higher pressures. In practice boilers are never fed at steam temperatures, so that the apparently obvious conclusion from the experiments that scale troubles are reduced with higher working pressures is as unjustified as it is contrary to experience. As regards the effect of scale on heat transference, the authors incline to the view that the losses in boiler efficiency owing to scale have been greatly exaggerated. They quote Eberle and Reutlinger as saying that their experiments showed that an ordinary scale averaging 0.22 inch in thickness influences the efficiency of a boiler so little that the effect cannot be determined by comparative boiler tests. With scales of medium conductivity, 0.2 inch thick, the loss of efficiency is considered to be only 2 or 3 per cent. In this connection it may be remarked that Messrs. Partridge and White found by experiment that the average coefficient of heat conductivity of calcium sulphate scale lay between 1 and 2 B.t.u.'s per square foot per hour per degree F. Although overall boiler efficiency may not be appreciably lowered by a moderate amount of scale, the overheating of the tubes may easily become serious. Taking the heat conductivity coefficient of sulphate scale at its average value of 1.5, a deposit only one hundredth of an inch thick will raise the temperature of a tube by about 5.6 degrees F. for every 10,000 B.t.u.'s per square foot per hour transmitted through the metal. With a heat transfer rate of 70,000 B.t.u.'s per square foot per hour, which is easily attained in tubes exposed to furnace radiation, a sulphate scale 0.1 inch thick would raise the temperature of a boiler tube working at 150 pounds gauge pressure to 753 degrees F. The same temperature would result from 0.07 inch of scale at 600 pounds pressure, and 0.04 inch at 1500 pounds, the rate of heat transfer being maintained at 70,000 B.t.u.'s per square foot per hour. As scales of lower conductivity would, of course, increase the tube temperatures, it is clear how little margin there is for scale in water-wall and similar tubes exposed to the intense radiant heat of a furnace.—"The Engineer," June 21, 1929.

BOILER EXPLOSIONS.

Boiler explosions are, fortunately, rare events in this country; so rare, indeed, that only those directly concerned with boilers, either constructors, inspectors or direct users, take such things into consideration. Nevertheless —or perhaps for that very reason—a recent report issued by the Board of Trade regarding an explosion which occurred about a year ago at an ironworks at Middlesbrough has attracted much attention and some critical comment. The subject is, however, in reality of much greater importance than the rare occurrence of an explosion would suggest, because quite a number of valuable and important boilers are from time to time put out of action by the discovery of defects which had not yet developed to the stage of final collapse. When all such cases are taken together they constitute a serious problem, although that problem has, fortunately, not yet assumed the degree of importance here which it had acquired a few years ago both in Germany and in America.

The case of failure reported upon by the Board of Trade Surveyor is clearly, from the published description, a typical example of what has come to be recognized as "inter-crystalline cracking at the rivet holes," and would, in America, have been confidently ascribed to "caustic embrittlement." The conclusion arrived at in the Board of Trade report is much more cautious than this, if indeed it can be called a conclusion at all. The

view is, at all events, expressed that the theory of "caustic embrittlement" is not accepted by the British boiler experts whom the Surveyor has consulted. While we disapprove of the term "caustic embrittlement" and of some of its implications, the rejection of the facts and ideas which lie at the base of that theory appears to be scarcely justified and may prove to some extent misleading in regard to British boiler practice.

The suggestion implied in the term "caustic embrittlement" that the action of caustic soda can render steel brittle is, of course, quite unsound. Even where steel has, probably under the action of caustic soda, developed a system of deep inter-crystalline cracks near the rivet holes of a boiler seam, the steel itself is not rendered brittle thereby. If a piece in which cracks are present is drawn out in a tensile testing machine there is no tendency for the cracks to spread further along the crystal junctions; each crack behaves exactly as would a similar incision produced by mechanical means, and the steel draws out in a perfectly ductile manner. Even under an impact test there is no evidence of notch brittleness unless one of the fine branching cracks has been included in the test piece and acts as an additional notch. The use of the term "embrittlement," therefore, is misleading and has probably made it difficult for some of our boiler experts to accept the whole theory because it appears to imply something—*i.e.*, general brittleness of the steel—which they know does not exist. On the other hand, the evidence obtained by American investigators, particularly by Parr and Straub, and to some extent confirmed by work at the National Physical Laboratory, definitely shows that inter-crystalline cracking is produced when caustic soda solutions of sufficient concentration act upon boiler plate steel which is at the same time exposed to heavy mechanical stress. Since it has yet to be shown that cracking of this definite and unusual type can be brought about in any other way which is even remotely related to boiler conditions, the implication is very strong that wherever in a boiler this type of cracking is encountered these two factors have probably been at work together. It is true that the concentration of soda which the American investigators have found necessary is absurdly high compared with what is found in boiler waters, but there is at least the possibility that local concentration in the narrow spaces between the steel plates in a riveted seam may occur under certain conditions. In the Middlesbrough case under discussion it is shown that the caulking was so carried out as to cause a slight bulge in one of the plates between two rows of rivets, and inside that bulge a space was provided in which, under the effects of slight leakage, concentration of the boiler water might go on.

The second difficulty which the "caustic cracking" theory—as we prefer to call it—has to face is that the operating stress required to produce cracking under the influence of the strong caustic solution is high—higher than the yield-stress of the steel. It is, of course, at once objected that the working stresses of a boiler are far below this limit, even where there are some stress concentrations between rivet holes. But it is clear that, in reality, other stresses than those due directly to the steam pressure must come into play. A seam between thick steel plates can only be kept steam-tight by the mutual elastic pressure of the two plates where they are in contact, and this is maintained by internal elastic stresses set up in the steel plates and in the rivets by the pressure used in closing the rivets and by their subsequent thermal contraction. The fact that many boilers successfully withstand the use of a certain amount of caustic in their feed water goes to show that in many cases existing boiler practice avoids the imposition of such stresses of undue severity, so that the elastic stresses required to keep a seam tight need not necessarily, even when added to the stresses due to

steam pressure, exceed the limit of safety in regard to the action of caustic soda. On the other hand, anyone familiar with the methods frequently used in riveting and caulking, or even with the appearance of many boiler seams, will admit that not infrequently very severe local internal stresses must be imposed on the steel. It would seem, therefore, that the conditions for "caustic cracking" as found by Parr and Straub may at times occur in boilers.

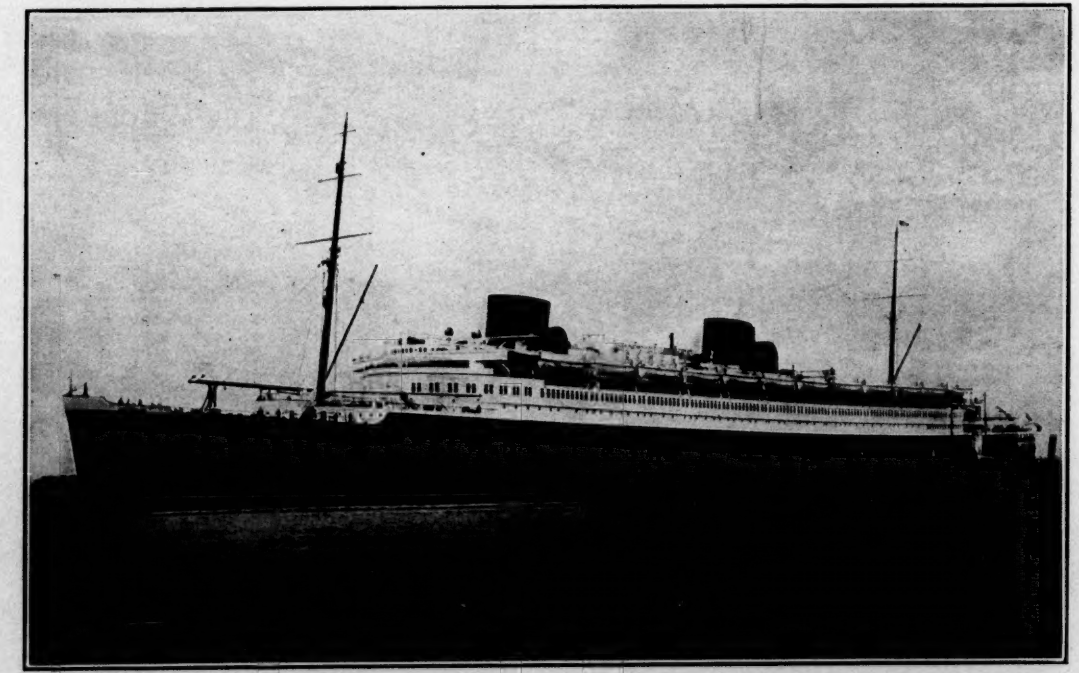
Whether or not this combination of conditions was responsible for the failure in the Middlesbrough case we must leave the reader to judge for himself from the published data. These show that caustic soda was present in the feed water, and the existence of serious stresses set up by riveting or caulking also appears to be indicated from such facts as the bulge in the plate referred to above. That no more definite conclusion was reached in the Board of Trade report may be due to the fact that the different experts entrusted with different parts of the investigation do not seem to have been given all the facts or placed in a position to form an opinion on the case as a whole. In itself, the absence of a definite conclusion from a report of this kind is not, perhaps, a very serious defect, but there is, in our opinion, no adequate emphasis on some of the most important factors. The really serious importance of avoiding entirely, if at all possible, the use of caustic soda or of chemicals likely to lead to its formation in boiler water is not emphasized in the recommendations, nor is any mention made of the fact that there is good reason to believe that additions of sulphate or, even better, phosphate of soda in quite small amounts tend to inhibit the destructive action even of concentrated soda. Much stress is laid upon the importance of inspection in locating the causes of leaky seams, but nothing is said of the dangerous effect of unduly high riveting pressures or of the risk that surrounds excessive caulking. The tendency to use severe caulking as a remedy for a leaky seam may well lead to an increased rate of destruction owing to the added stresses locally imposed on the steel.

In making these remarks we do not desire to be critical of the Board of Trade Surveyors; the whole matter is one of admittedly great difficulty, and they are not to be blamed for taking up a cautious attitude with regard to the theory of "caustic cracking," but the importance of the issues involved is so great that a thorough inquiry seems to be called for. Possibly further experimental investigation is needed to test the accuracy of the conclusions of Parr and Straub and their applicability to practical boiler conditions. In any event, however, engineers cannot afford to remain in a state of uncertainty as to the soundness of their practice in regard either to boiler construction or working.—"Supplement to The Engineer," May 31, 1929.

THE *BREMEN*.

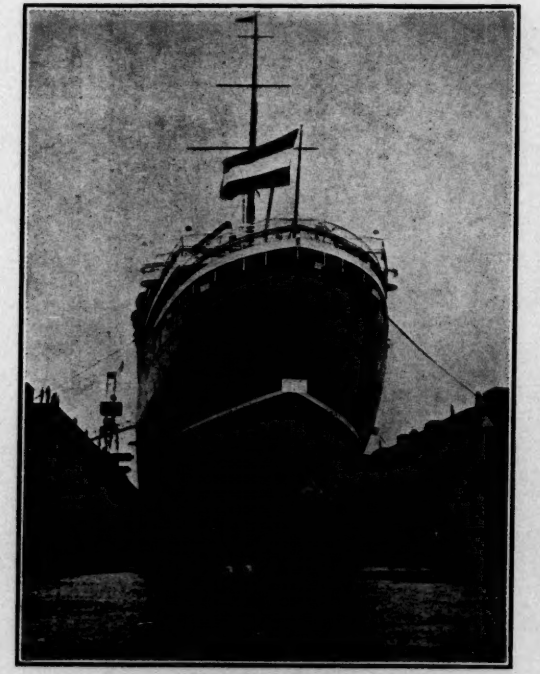
The Norddeutscher Lloyd liner *Bremen* arrived at Southampton from Bremen on Monday, and, drawing about 30 feet of water, was docked on the large floating dock by Harland & Wolff, Ltd., who are cleaning and repainting the hull preparatory to the progressive trials which are to be carried out in the North Sea during the next few days.

The photograph of the ship coming up Southampton Water shows her strange profile, with two widely-spaced squat funnels more reminiscent of such motorships as the *Asturias* than a high-powered steamship burning oil fuel. The promenade decks are well covered in with their sliding saloon windows. The large lifeboats, each accommodating 145 persons, loftily

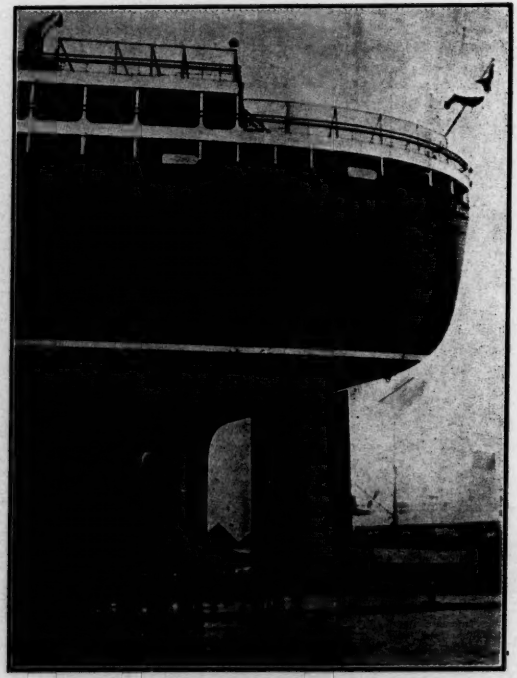


THE "BREMEN," OF ABOUT 46,000 GROSS TONS, COMING INTO SOUTHAMPTON. NOTE THE WIDE SPACING OF THE FUNNELS, THE AEROPLANE CATAPULT-LAUNCHING DEVICE BETWEEN THEM, AND THE EXTENSIVE SPORTS DECK AREA BEHIND THE AFTER FUNNEL.

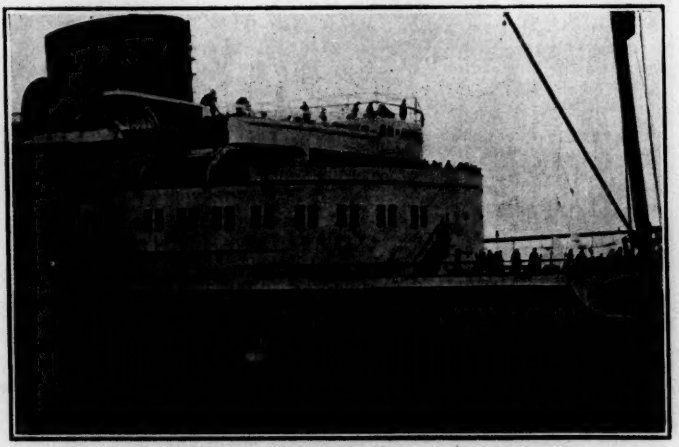
BOW VIEW OF THE "BREMEN."



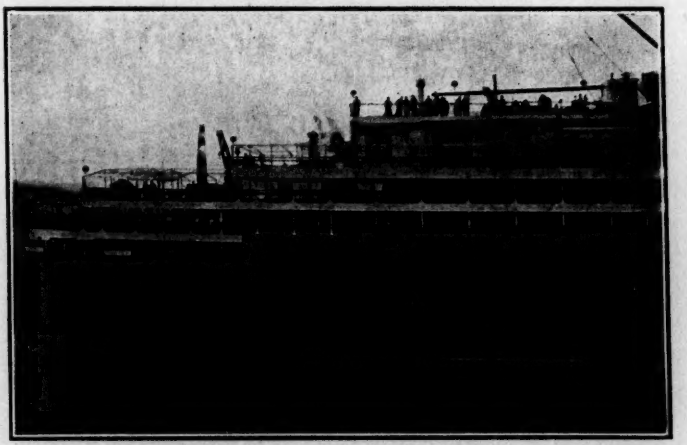
VIEW OF STERN SHOWING POSITION OF PROPELLERS AND THE FINE LINES AFT.



PROFILE VIEW OF STERN, SHOWING STREAM-LINE RUDDER AND PITCH OF PROPELLERS.



General View of Passenger Decks Forward, Showing Curved Sweep of the Superstructure.



Stern View, Showing Arrangement and Height of Passenger Decks Aft.

poised on Welin Maclachlan davits of exceptional size, increase the apparent height of the superstructure and, above, between the two funnels, is a huge girder construction, the catapult device for launching aeroplanes.

On the voyage from Bremerhaven, foul as she must have been, she is said to have averaged 26.6 knots for three hours with ease, and for an hour to have done 28 knots.

The circumstances under which the large dock at Hamburg is at present occupied with her sister ship, the *Europa*, are well known, and the fact that one of these much-discussed ships was on view, high and dry, from the town which claims to be the home of all the British mammoths, and is a place of call for so many of the liners of other nations, naturally attracted a large crowd of spectators and local experts.

In addition, by a very gracious act on the part of her owners, formal invitations were extended to leading British shipowners, naval architects and engineers to inspect the whole of the vessel during her short, and one may almost say informal, visit before emerging into the fuller public glare.

In many ways she is a distinguished vessel. Built to accomplish the voyage between Bremerhaven and New York in six days, she must maintain a high average speed, so great that inevitably in regular service, she and her sister ship must be the fastest ships in the world, notwithstanding the probability that the *Mauretania* on occasion may still show them a clean pair of heels. But no one regarding that fine and pecular hull form with its carefully designed lines, the absence of bilge keels and an apparent block coefficient of 0.55 can doubt that the ships are built for speed and, under suitable weather conditions, will make it every time.

To the large group of sightseers watching from the steps of the Hythe Ferry, the curious bulbous bow is the center of interest, as indeed it was at the launch less than twelve months ago. Towering above is the enormous superstructure, its graceful curves forward accentuating the straightness of the bridge with its exceptional overhang. It seemed difficult to the lay observers of the crowd that such a fine underbody could adequately support in equilibrium the upper decks with their elaborate accommodation unsurpassed in any ship.

Those who took the ferry or one of the many craft plying to show the other end of the dock, saw at the stern much of greater interest. Here the fineness is more evident, and in no way is it shown more plainly than in the distance of the propeller shafts above the keel line, an obvious indication of the height of the engine-room floors.

Evidently as a compromise and in order to give adequate submergence to the screws in the 35 feet which is, apparently, the normal draught of the ship, propellers of small diameter and good pitch have been fitted, suggesting that they will revolve in the region of 300 revolutions per minute so as to maintain about 27 knots, with an estimated 120,000 S.H.P.

Beyond doubt the Norddeutscher Lloyd has created two epoch-making vessels, and, looking at the *Bremen* in dock, the impression it conveys is that considerable courage was required to introduce the design. It becomes evident, when looking at the form, that there is still great scope for naval architects provided that shipowners are prepared boldly to overthrow conventional ideas and to disregard traditional requirements.—"Shipbuilding and Shipping Record," July 4, 1929.

THE M.A.N. DOUBLE-ACTING TWO-STROKE CYCLE MARINE DIESEL ENGINE.

By HEINRICH BECKER.*

(A) GENERAL: COMPARISONS OF SPACE AND WEIGHT.

The construction of Diesel engines has undergone a change within the last few years in view of the fact that those firms engaged in the building of such engines are now replacing the single-acting—which is, more or less, at the limit of its output—with the double-acting engine. A few years ago well-known scientists and experts stated that the double-acting two-stroke cycle engine was bound to come and that its introduction was only a question of time.

The four-stroke cycle Diesel engine practically predominated for the first 15 to 20 years after its appearance on the market. At the present time the Maschinenfabrik Augsburg-Nürnberg A.-G. and several of their licensees are still constructing single-acting four-stroke cycle engines for installation in ships, and these have been considerably improved. In this way the needs of individual shipowners—such, for example, as the Scandinavian owners—have been taken into consideration. A large number of these engines have been, and are still being, constructed by the M.A.N. Swedish licensees, the firm Kockums, of Malmö, *i.e.*, engines of approximately 3000 E.H.P. At the present time two compressorless engines of this type each of 3600 E.H.P. are being built at the Augsburg Works, for account of Spanish owners.

In addition to these four-stroke cycle engines, single-acting two-stroke cycle engines up to 7000 E.H.P. are being built by the M.A.N. and their licensees.

As time went on, an important change in favor of the double-acting two-stroke cycle engine took place, and this type is being built by M.A.N. licensees in most of the maritime countries. Figure 1 shows the development of this type of engine.

A similar development to that of the Diesel engine is also noticeable in the case of piston steam engines and gas engines. Both types were originally introduced as single-acting engines, to be superseded by double-acting engines. Today it is almost impossible to purchase or to have constructed single-acting steam or large gas engines.

If one wishes to build the Diesel motor as a two-stroke cycle engine, the following fundamental conditions must be fulfilled in order to secure efficient scavenging:—

(1) The exhaust gases must be expelled from the cylinder completely; and

(2) The combustion space must be filled with an adequate supply of fresh air for the subsequent combustion with the least possible expenditure of work.

Condition (1) can be carried out with the opposed-piston type of engine and with the valve-scavenging engine; but in the case of the former this is very costly on account of the extensive mechanism, and the latter is at a disadvantage owing to the valves being fitted in the cylinder cover and because of the valve gear necessary.

Condition (2) cannot be fulfilled satisfactorily with the valve-scavenging engine. The fact that the scavenging valves are in the cylinder cover gives rise to difficulties on account of the unfavorable shape of the cover, and the area of opening which can be secured in the scavenging valves of fast-

* Chief Engineer at the Augsburg Works of the M.A.N.

running engines is comparatively small. The scavenging air must, therefore, have a much higher pressure—a fact which necessitates the expenditure of much more work at the scavenge pump, and also means still more loss. As an example, in fast-running engines, with efficient port scavenging and the scavenge pump working at about 6 per cent of the Indicated H.P., this would be increased to approximately 10 to 12 per cent in the case of valve scavenging.

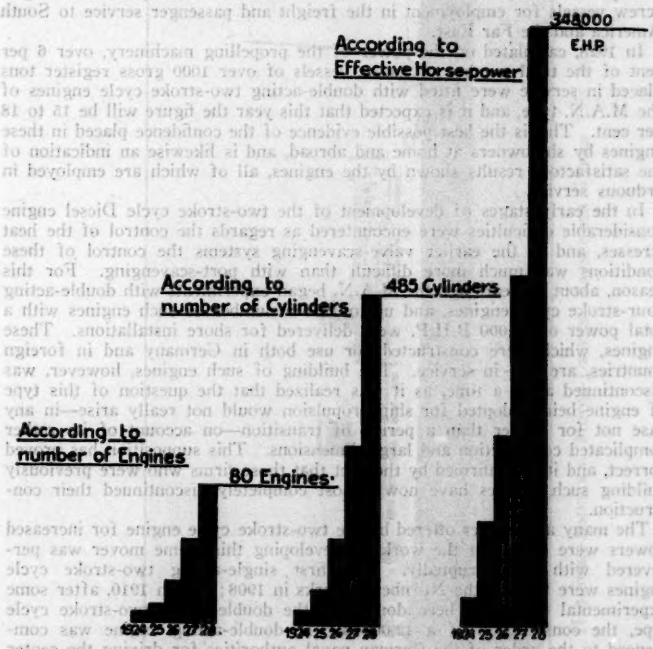


FIG. 1.—DEVELOPMENT IN THE CONSTRUCTION OF M.A.N. DOUBLE-ACTING TWO-STROKE CYCLE DIESEL ENGINES.

The double-acting two-stroke cycle Diesel engine depends especially on the fulfilment of the conditions named for the lower end of the cylinder, which, on account of the piston rod, involves entirely different conditions for expelling the exhaust gases by means of the scavenging air. If these conditions can be fulfilled—and they are most difficult—combustion can easily be obtained by well-known means. It is generally known that all the important builders of Diesel engines have been engaged for a good many years in an effort to find some solution. The attempts have so far been mostly unsuccessful, or at least no definite results have been achieved.

Several years ago, after extended experiments, the M.A.N. were successful in finding the best means of scavenging, which is similar to that ob-

tained in the opposed-piston engine. In this manner the foundation was laid for the practical double-acting two-stroke cycle Diesel motor, which has been extensively adopted in a comparatively short time. Reports published in 1928 of ships delivered and their method of propulsion show how remarkable is the increase in the number of ships fitted with double-acting two-stroke cycle Diesel engines of the M.A.N. type. Of 28 motor vessels delivered in Germany during this period, seven had engines of this type with an aggregate power of 32,300 B.H.P. and 38,850 I.H.P. These ships are all single-screw vessels for employment in the freight and passenger service to South America and the Far East.

In 1928, calculated on the power of the propelling machinery, over 6 per cent of the total number of motor vessels of over 1000 gross register tons placed in service were fitted with double-acting two-stroke cycle engines of the M.A.N. type, and it is expected that this year the figure will be 15 to 18 per cent. This is the best possible evidence of the confidence placed in these engines by shipowners at home and abroad, and is likewise an indication of the satisfactory results shown by the engines, all of which are employed in arduous service.

In the early stages of development of the two-stroke cycle Diesel engine considerable difficulties were encountered as regards the control of the heat stresses, and in the earlier valve-scavenging systems the control of these conditions was much more difficult than with port-scavenging. For this reason, about 20 years ago the M.A.N. began experiments with double-acting four-stroke cycle engines, and up to 1912 a number of such engines with a total power of 24,000 B.H.P. were delivered for shore installations. These engines, which were constructed for use both in Germany and in foreign countries, are still in service. The building of such engines, however, was discontinued after a time, as it was realized that the question of this type of engine being adopted for ship propulsion would not really arise—in any case not for longer than a period of transition—on account of its rather complicated construction and large dimensions. This supposition has proved correct, and it is confirmed by the fact that those firms who were previously building such engines have now almost completely discontinued their construction.

The many advantages offered by the two-stroke cycle engine for increased powers were such that the work of developing this prime mover was persevered with uninterruptedly. The first single-acting two-stroke cycle engines were built at the Nürnberg Works in 1908; and in 1910, after some experimental work had been done with the double-acting two-stroke cycle type, the construction of a 12,000-B.H.P. double-acting engine was commenced to the order of the German naval authorities for driving the center propeller of a battleship. This engine was designed as a valve-scavenging engine, and was delivered in 1917 after a trial run lasting five days. Complete success was achieved with this engine, which had a cylinder diameter of 850 millimeters and a stroke of 1050 millimeters and ran at 160 revolutions per minute. The engine could not only be driven at the power named but could also be overloaded, and the very strict conditions specified at that time were entirely fulfilled.

Efforts were then made to develop a so-called port-scavenging system in place of the valve scavenge, in order to attain, by dispensing with the valves, improved working safety, and a higher degree of efficiency, service and production. The experiments extending over some years which were necessary in this connection resulted, in 1920, in the now well-known M.A.N. system of port-scavenging, and this proved entirely practical on several trial engines.

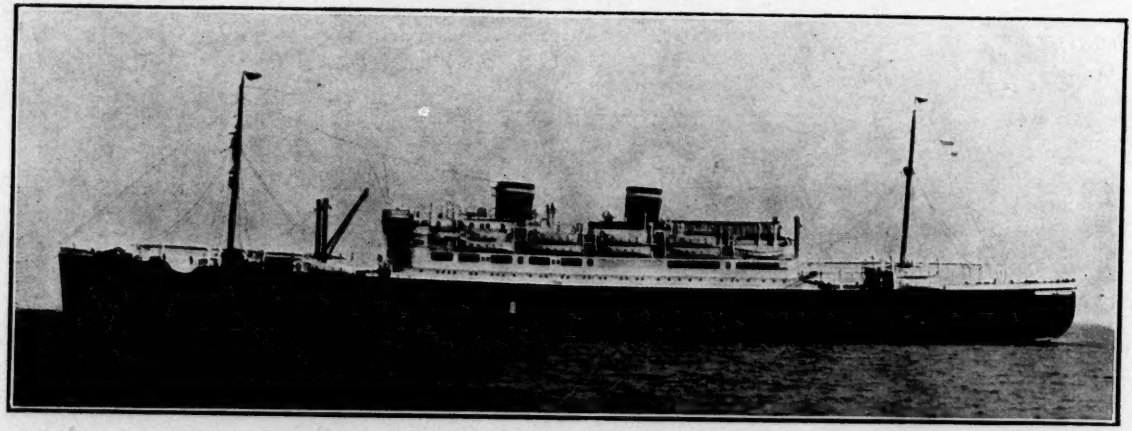


FIG. 2.—THE HAMBURG-AMERIKA MOTOR-DRIVEN PASSENGER LINER "ST. LOUIS."

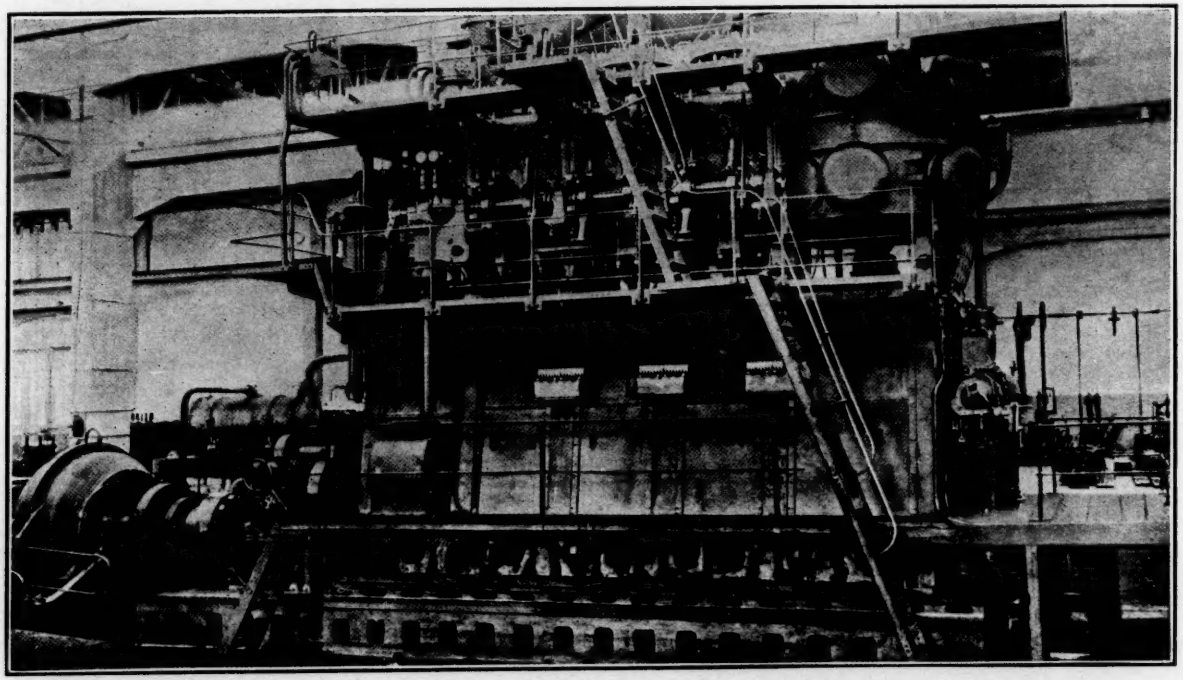


FIG. 3.—M.A.N.-SAVOIA DOUBLE-ACTING TWO-STROKE CYCLE THREE-CYLINDER DIESEL ENGINE, 1500 B.H.P. AT 190 R.P.M., WITH COMPRESSOR AND SCAVENGING PUMP.

This scavenging system laid the foundation for the success of the double-acting two-stroke cycle engine. The characteristics of the M.A.N. scavenging system, which have frequently been described in the technical press, are generally known. The scavenging and exhaust ports are on the same side of the cylinder; the scavenging air on entering is diverted at the piston bottom, sweeps the wall of the cylinder lying opposite, and then again returns across the cylinder cover and flows down through the exhaust ports. It is important here to note that the scavenging air is directed through the entire cylinder, and in this respect differs materially from other scavenging systems. Owing to the fact that the scavenging air drives the exhaust gas before it, only the minimum quantity of air is required. This system is also specially well adapted for the efficient scavenging of the lower cylinder ends of double-acting engines. As evidence of the efficiency of scavenging, it may be stated that it is possible to attain a mean indicated pressure of about 8 kilograms per square centimeter with good exhaust and without supercharging.

This mean pressure has also been attained with *fast-running* double-acting engines—for instance, with an engine having a cylinder power of 900 B.H.P. at 300 revolutions per minute. Under normal running conditions an indicated pressure of about 5.7 to 6 kilograms per square centimeter can be realized with the greatest safety and with entire satisfaction as regards good combustion, durability of parts subject to wear and tear, and absence of excessive heat stresses. This system of scavenging is not only suitable for engines of large dimensions—which is confirmed by the performance of the 15,000-B.H.P. engine installed in Hamburg Neuhof Power Station—but also for fast-running engines, as well as those of small and even the smallest dimensions. The double-acting engine is absolutely reliable and dependable, with good and simple scavenging and good combustion; and all these conditions are entirely fulfilled by the M.A.N. engine. On account of many years' experience with large gas engines and double-acting Diesel motors, mechanical difficulties have long since been wholly overcome. The engines delivered during the last few years work with a consumption of scavenging air equal to 1.35 times the swept volume of the working cylinder, as against the 1.65 to 2 necessary in the case of other scavenging methods. This low consumption of air has not been attained by any other method of scavenging.

The figures given in Table I relating to the driving power of the turbo-blowers and the scavenging-air pressures are mean values taken from the

TABLE I.

Motorships.	I.H.P./B.H.P. of main engine.	Cylinder diameter and stroke, mm.	No. of Cylinders	Revs. per min.	Scavenging air pressure mm. water col.	Power of blowers, H.P.
San Francisco	4500/3750	700/1200	5	87	800	130
Trave	5500/4600	700/1200	6	82	900	165
Burgenland	6000/5000	700/1200	6	95	1000	228
Seattle	6400/5400	700/1200	7	85	1000	204
Palatia	3800/3200	600/900	6	120	850	153

FIG. 3.—M.A.N.-SAVOIA DOUBLE-ACTING TWO-STROKE CYCLE THREE-CYLINDER DIESEL ENGINE, 1800 B.H.P. AT 150 R.P.M., WITH COMPRESSOR AND SCAVENGING PUMP.

first voyages of the ships mentioned. These have been selected from the large number of ships fitted with double-acting two-stroke cycle engines now in service.

The power expended for scavenging the working cylinder, expressed as a ratio of the Indicated H.P. of the main engine, is therefore as follows:—

San Francisco, per cent	2.9
Trave, per cent	3.0
Burgenland, per cent	3.8
Seattle, per cent	3.2
Palatia, per cent	4.0

Of the ships in service, the largest passenger vessel with Diesel engines is the *Augustus*, owned by the Navigazione Generale Italiana, which was placed in service in October, 1927, and is fitted with four double-acting two-stroke cycle engines with a total output of 28,000 B.H.P. After the piston rods had been changed for rods of Siemens-Martin steel with a rearrangement of the cooling-water inlet and outlet, carried out after the second voyage, each trip has been made according to schedule and without the slightest inconvenience. The main engines are driven at an average speed of 119 revolutions per minute and thus give a normal power of 25,000 B.H.P. The silence and freedom from vibration of the engines are especially remarkable.

Notable new motorships under construction during the current year are the passenger liners *St. Louis* (Figure 2) and *Milwaukee*, for the Hamburg-Amerika Line. The first-named of these two ships has already been placed in the North Atlantic service, having been delivered from the Bremer Vulkan Works on the 22nd of March. Her speed is 16.5 knots and she has a gross register tonnage of 16,772. The four propelling engines with a total power of 12,200 B.H.P., two connected to each of the two propeller shafts, have been built for the first time as fast-running double-acting two-stroke cycle engines working at 225 revolutions per minute, and by means of mechanical reduction gearing the propeller revolutions are reduced to 110 per minute. The Vulcan couplings enable one engine to be shut off when working full speed by emptying the oil clutch; the engine can likewise be started again by filling up the clutch. Corresponding tests were carried out during the trial trip.

The sister ship *Milwaukee*, under construction by Messrs. Blohm & Voss, will be ready shortly. Her machinery installation differs from that of the *St. Louis* in that the engines are coupled directly to the reduction gearing. Both installations have the advantage of low height of main engines, thus providing increased space on the decks above for the accommodation of passengers.

Two passenger ships have recently been built in Italy—the *Olbia* and *Attilio Deffenu*—and these are each fitted with two M.A.N. engines of 1500 B.H.P. This installation is especially interesting owing to the fact that the main engines have for the first time been built as three-cylinder units; see Figure 3. The scavenging pump and compressor are built on the engine, and the length of the engine is small. There was no disturbing vibration in spite of the three-cylinder arrangement.

The question of increased power has again come into prominence during recent years, and installations up to 180,000 B.H.P. have been considered. For this power, engines working at high revolution are required. To meet this condition the double-acting two-stroke cycle engine is suitable on account of its low specific weight and the small space required. Whether engines coupled directly or through mechanical reduction gears, with or without

liquid clutches, will find extensive use for such powers cannot be stated at present. The engine coupled directly takes up more space on account of the low speed of rotation; while with mechanical reduction gearing fast-running engines could be adopted, which would allow for economy in height. The following notes give particulars of a few installations.

Figure 4 shows the engine installation of a French passenger liner under construction, with four directly coupled main engines each of 4500 B.H.P.;

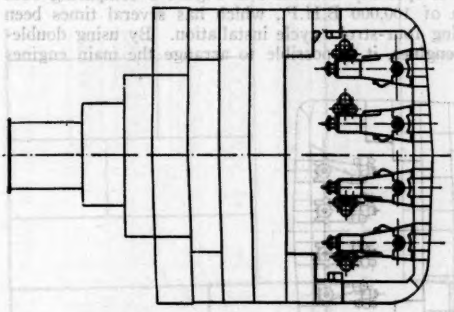
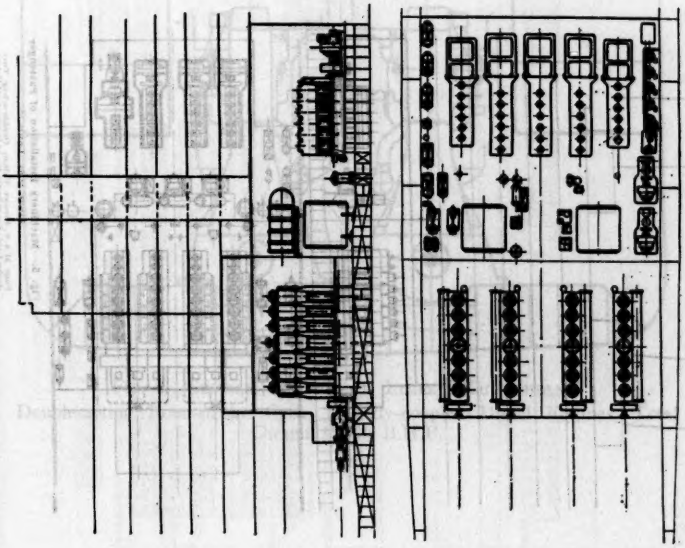


Fig. 4.—Machinery Installation of French Passenger Liner under Construction.

Four M.A.N. Double-acting Two-stroke Cycle Diesel Engines. Total Output 18,000 B.H.P.



while Figure 5 illustrates the engine installation of the *St. Louis*, which, as already mentioned, is fitted with M.A.N fast-running double-acting two-stroke cycle engines with intermediate connecting mechanical gearing and liquid clutch. As a comparison, Figure 6 gives a transverse sectional drawing of the motor liner *Kungsholm* with double-acting four-stroke cycle engines. The three last-named illustrations show the valuable economy in engine space to be gained by using the double-acting two-stroke cycle engine as against the double-acting four-stroke cycle type.

The economy in space is specially noticeable in Figure 8 comparing both types for an installation of 100,000 B.H.P., which has several times been stated to be a double-acting four-stroke cycle installation. By using double-acting two-stroke cycle engines, it is possible to arrange the main engines

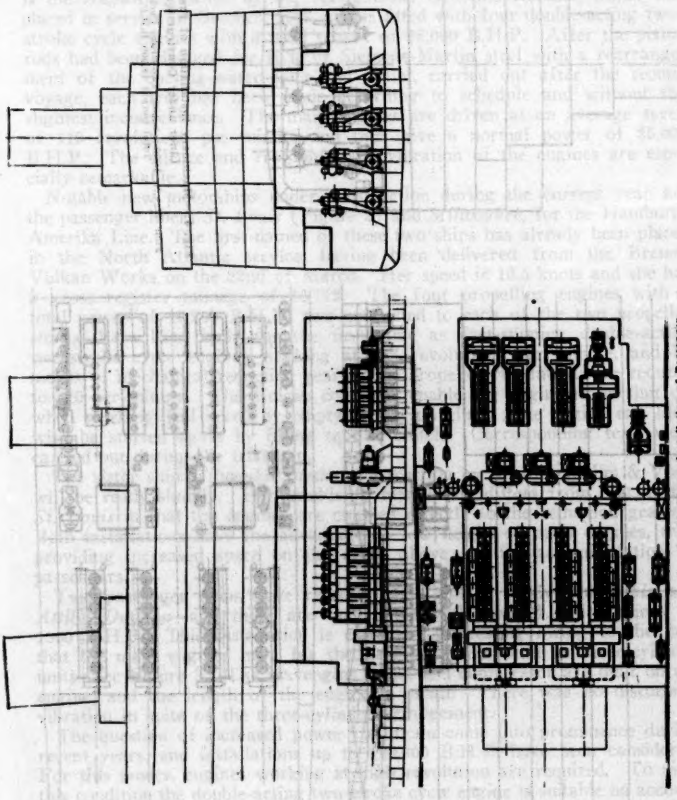


Fig. 5.—Machinery Installation of Passenger Liner "St. Louis."
Four M.A.N. Reversing Valveless Double-acting Two-stroke Cycle Single-cylinder Diesel Engines with Mechanical Gear. Total Output 17,000 H.P. at 225 r.p.m.

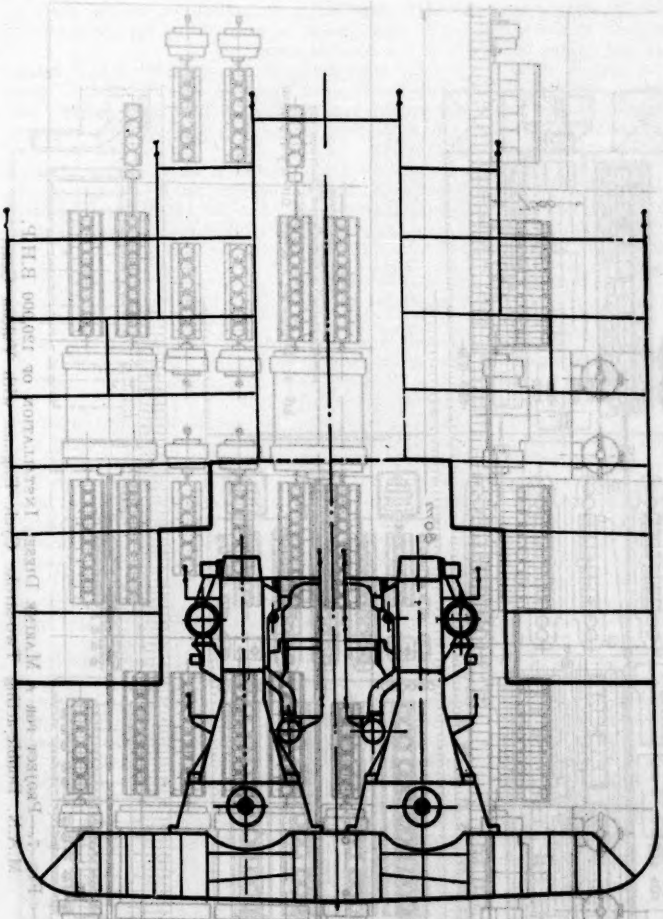


FIG. 6.—SECTION OF SWEDISH LINER "KUNGSHOLM."
Double-acting Four-stroke Cycle Directly-coupled Diesel Engines, Total Output 15,000 B.H.P.

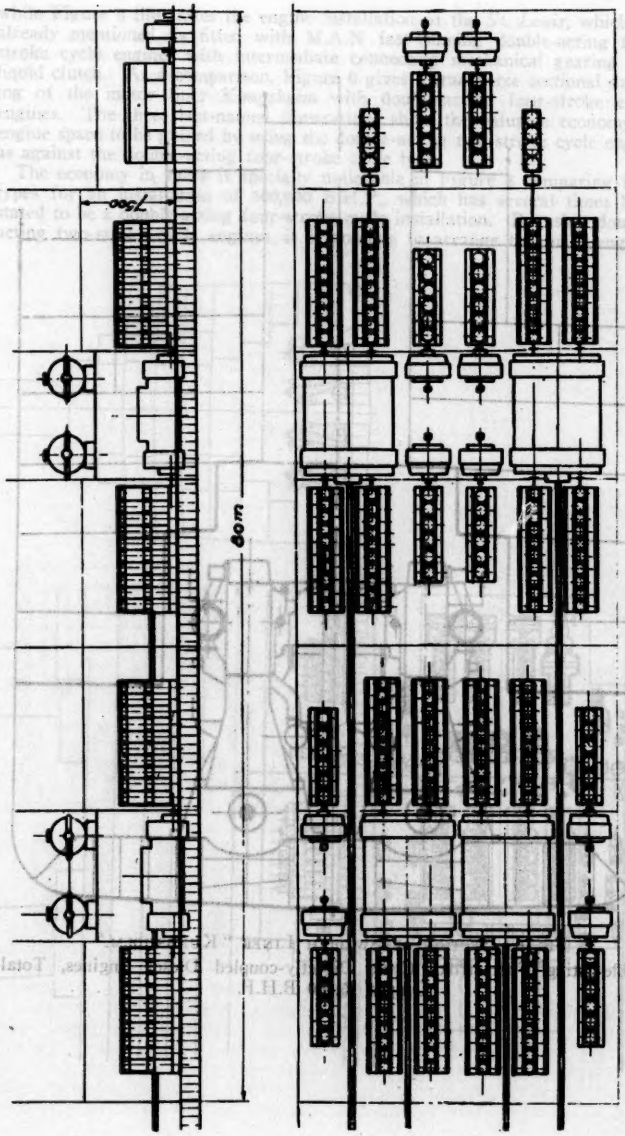


FIG. 7.—PROJECT FOR A MARINE DIESEL INSTALLATION OF 120,000 B.H.P.
M.A.N. Double-acting Two-stroke Cycle Engines with Vulcan Gear.

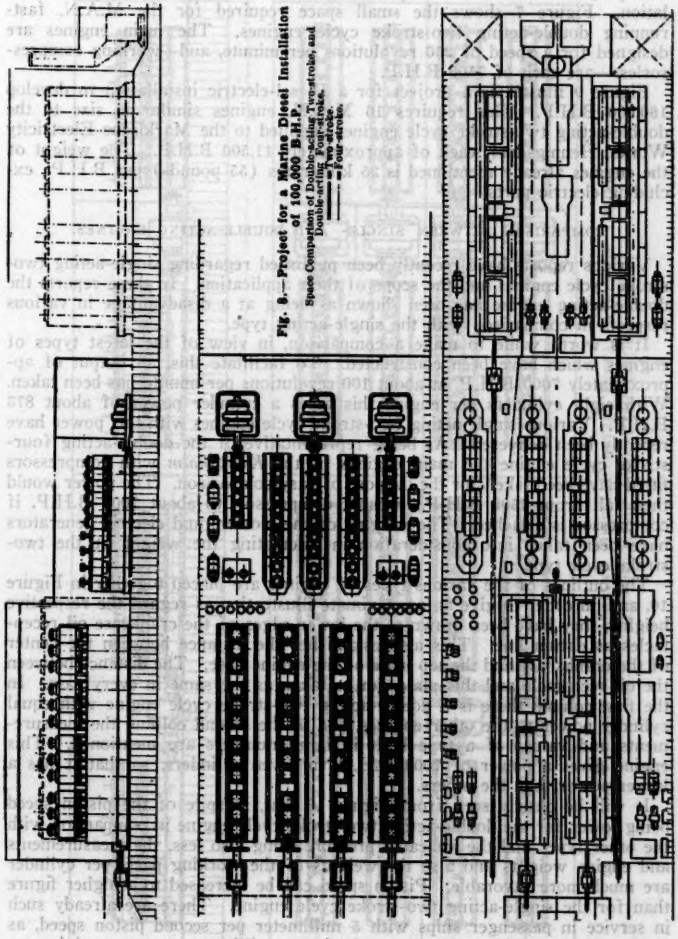


Fig. 8.—Project for a Marine Diesel Installation of 100,000 B.H.P.

Space Comparison of Double-acting Two-stroke and Single-acting Four-stroke.

——— = Two-stroke.

——— = Four-stroke.

side by side, on account of the small piston stroke and the smaller width of engine. In this way a neater arrangement is obtained and approximately 40 per cent saving is effected in engine-room space.

A project which has recently been studied is for a 120,000-B.H.P. installation. Figure 7 shows the small space required for the M.A.N. fast-running double-acting two-stroke cycle engines. The main engines are designed for a speed of 250 revolutions per minute, and—working compressorless—are each of 7500 B.H.P.

Figure 9 illustrates a project for a Diesel-electric installation to develop 180,000 B.H.P. This requires 16 M.A.N. engines similar in size to the double-acting two-stroke cycle engines supplied to the Markische Electricity Works, Hennigsdorf, each of approximately 11,500 B.H.P. The weight of the engines already mentioned is 25 kilograms (55 pounds) per B.H.P., excluding electric parts.

COMPARISON BETWEEN SINGLE- AND DOUBLE-ACTING ENGINES.

Various reports have recently been published regarding single-acting two-stroke cycle engines and the scope of their application. In these reports the double-acting engine has been shown as being at a disadvantage in various respects in comparison with the single-acting type.

It is worth while to make a comparison, in view of the latest types of engines which have been constructed. To facilitate this, an output of approximately 7000 B.H.P. at about 100 revolutions per minute has been taken. With eight cylinders per engine this gives a cylinder power of about 875 B.H.P. Various single-acting two-stroke cycle engines with this power have recently been delivered. As being representative of the double-acting four-stroke cycle engine the main engines of the *Kungsholm* with compressors fitted have been taken for the purpose of easy comparison. The power would then fall from 7500 B.H.P., without compressor, to about 7000 B.H.P. if compressor is attached. The weights of the blowers and electric generators have been taken into consideration in calculating the weight of the two-stroke cycle type.

The outlines of the various types of engines are placed together in Figure 10, and, in order to give an approximate illustration as regard the respective heights, they have been drawn to the lower edges of the crankcase oil receptacles as datum line. This also establishes the distance between the center of the crankshaft and the top plate of the engine base. The distance between the oil receptacle and this plate can be taken as the same in every case. In the first column there is a double-acting two-stroke cycle engine with equal cylinder power to the other engines, and in the second column the measurements and values of a type built in large numbers are mentioned. This engine gives a power of 7000 B.H.P. in seven cylinders, so that it has a larger power than the others.

It will be clearly seen from Figure 10 that, in spite of the piston speed being lower for the double-acting two-stroke cycle engine in comparison with the other types and the indicated pressure being also less, the measurements and engine weights and also the weights of the working parts per cylinder are much more favorable. Piston speed can be increased to a higher figure than for the single-acting two-stroke cycle engine. There are already such in service in passenger ships with 5 millimeter per second piston speed, as well as engines with a piston speed of 6.45 millimeters per second in stationary installations. If in the case of ships' engines the piston speed of 4.2 millimeters per second is maintained, this is accounted for by the fact that the number of revolutions was fixed and there was really no necessity for it to be increased.

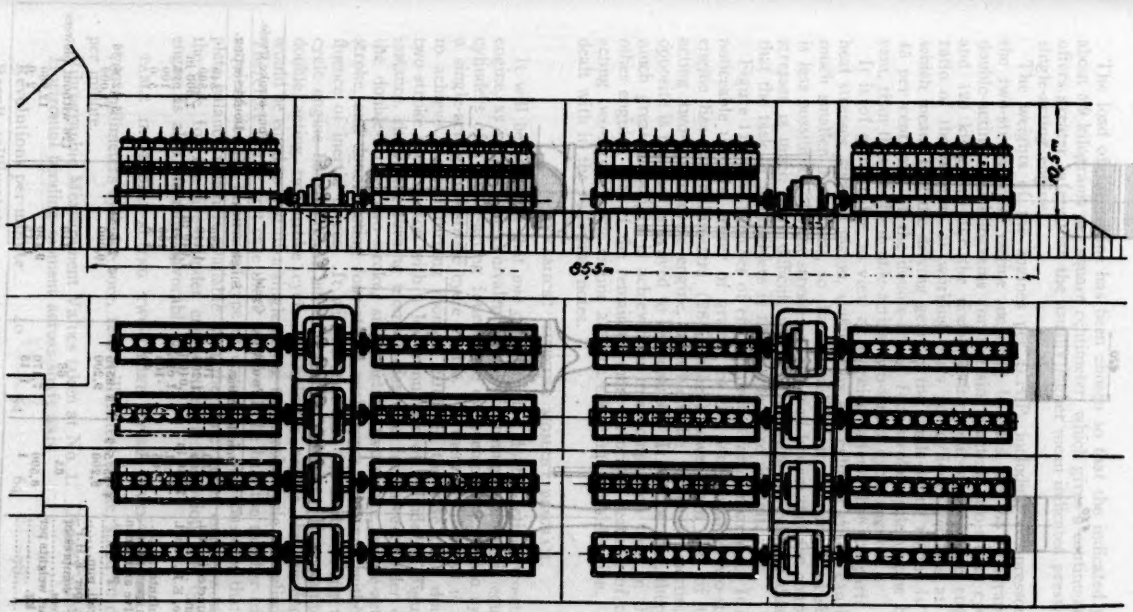
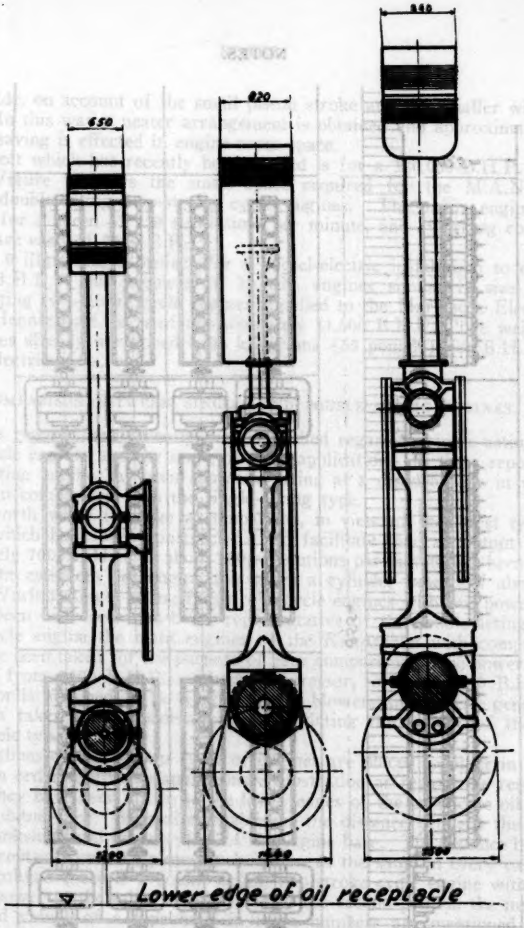


FIG. 9.—PROJECT FOR A MARINE DIESEL-ELECTRIC INSTALLATION OF 180,000 B.H.P.
M.A.N. Double-acting Two-stroke Cycle Engines.



	Two-stroke Cycle.			Four-stroke Cycle.
	Double-acting.		Single-acting.	Double-acting.
Cylinder diameter in mm.	650	700	820	840
Stroke in mm.	1,200	1,200	1,440	1,500
Performance, E.H.P. cyl.	7,040 in 8 cyl.	7,040 in 7 cyl.	7,040 in 8 cyl.	7,000 in 8 cyl.
Revs. per minute	105	105	100	100
Mean piston speed m./sec.	4·2	4·2	4·8	5·0
Mean effective and mean indicated pressures, kg./cm.²	4·95/5·9	4·95/5·9	5·2/6·34	5·12/6·56
Height overall, mm.	9,200	9,200	10,000	11,000
Weight, kg. per E.H.P., including compressor and blowers	65	66	110	126 without blowers
Working-part weights per cylinder, kg.	6,500	7,470	9,430	12,960
Ratio	1	1·15	1·45	2·0

FIG. 10.—COMPARISON OF THE CHARACTERISTISC OF VARIOUS MARINE DIESEL ENGINES.

The load of the engine has been chosen so that the indicated pressure is about 5.9 kilograms per square centimeter, which gives a continuous load and offers greater safety than the usually higher mean indicated pressure for the single-acting engine.

The weights of the engines per B.H.P. including compressor, and for the two-stroke cycle engine including blowers, are 65 kilograms for the double-acting, 110 kilograms for the single-acting two-stroke cycle engine, and 126 kilograms for the double-acting four-stroke cycle engine. The ratio of the weights of working parts per cylinder remains at 1:1.45:2.0, which means that the driving gear of the single-acting engine is heavier by 45 per cent, and of the double-acting four-stroke cycle engine by 100 per cent, than that of the double-acting two-stroke cycle engine.

It is of great value, in view of working safety, that the parts subject to heat stresses, such as piston, stuffing box and cylinder cover, can be built of much smaller dimensions, so that, in addition to the simplified shape, there is less possibility of heat stresses. It is well known that the control of heat stresses is much more difficult with the increase of cylinder diameter, and that the task of the builders is thus made more difficult.

Figure 11 shows sketches of engines drawn for comparison. It is specially noticeable that the center of gravity of the dougle-acting two-stroke cycle engine lies 730 millimeters (28¾ inches) lower than that of the single-acting two-stroke cycle engine, by which the frequent assertion that the opposite is the case is proved to be incorrect. It will be seen, therefore, that much greater stability is achieved in this comparison than in the case of other engines, even if the remaining more favorable features of the double-acting two-stroke cycle engine are not taken into consideration. These are dealt with in the following notes.

(B) DYNAMICS: TURNING MOMENT DIAGRAM.

It will be apparent that *one* cylinder of a double-acting two-stroke cycle engine, as regards the equivalent range of turning moment, is equal to *four* cylinders of a single-acting four-stroke cycle engine or to two cylinders of a single-acting two-stroke cycle engine. It therefore follows it is possible to achieve a given turning moment effect in the case of a double-acting two-stroke cycle motor with a lower number of cylinders. Figure 12, for instance, shows the turning moment diagrams for three-cylinder engines of the double-acting two-stroke, single-acting two-stroke, single-acting four-stroke, and double-acting four-stroke cycle types, with and without the influence of inertia forces. It will be seen that the double-acting two-stroke cycle engine has the most uniform turning moment. For example, with a double acting two-stroke cycle five-cylinder engine the turning moment would be equal to that of a single-acting two-stroke cycle ten-cylinder engine. Therefore in the first case there would only be half the number of cylinders required for the last-named type of engine. If one considers that the complete balance which is available with the ten-cylinder engine would also be the case for the five-cylinder engine, then it is obvious the double-acting engine is also the most favorable here.

TABLE II.—DOUBLE-ACTING TWO-STROKE CYCLE SEVEN-CYLINDER MOTOR.

700 millimeters cylinder bore, 1200 millimeters stroke, speed 90 revolutions per minute.

Vibrographic Measurement Values taken at No. 1. Upper Cylinder Cover. Horizontal bending moment across shaft axis.

Revolutions per minute	40	54	64	78	89
Result, millimeters	±0.14	±0.20	±0.14	±0.14	±0.14

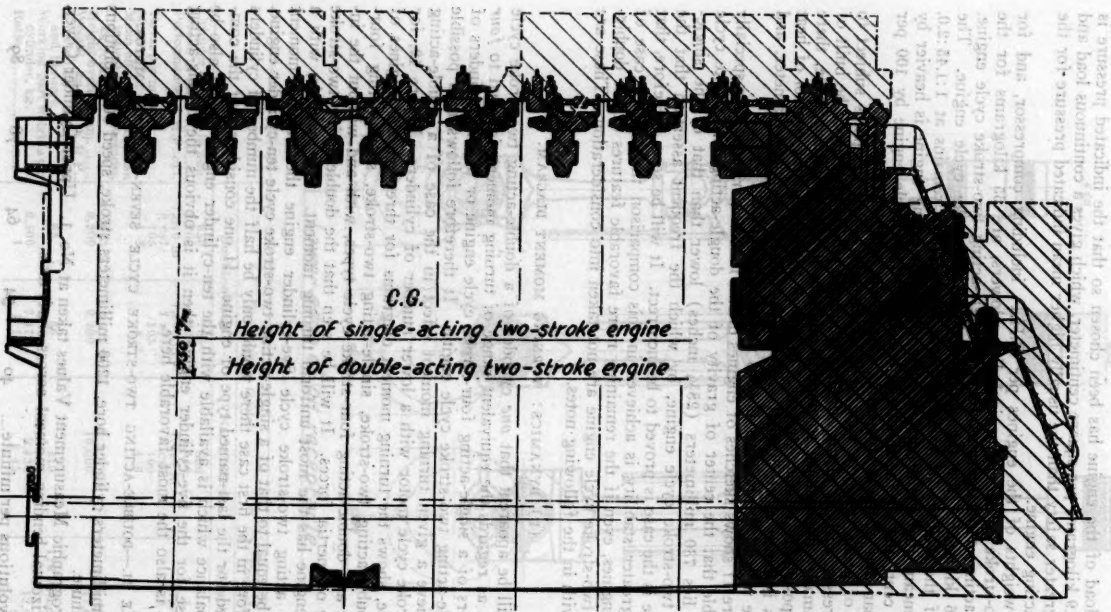
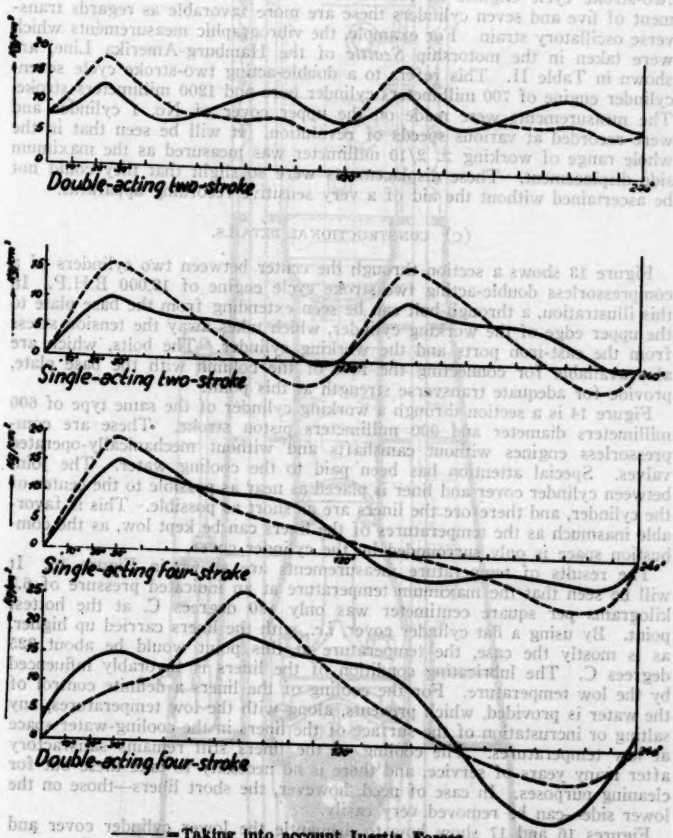


Fig. 11.—Comparison of Space required for various Marine Diesel Engines.

——— = Double-acting Two-stroke, 7,040 B.H.P. at 105 r.p.m.
——— = Single-acting Two-stroke 7,040 B.H.P. at 100 r.p.m.
—·—·— = Double-acting Four-stroke, 7,000 B.H.P. at 100 r.p.m.



——— = Taking into account Inertia Forces.
– – – – – = Excluding Inertia Forces.

Fig. 12.—Turning Moment Diagrams of various Three-cylinder Engines.

Crank Displacement = 120°.

Experience and careful vibrographic tests have proved that no noticeable vibration could be detected in the numerous installations of double-acting two-stroke cycle engines so far constructed, and in the unusual arrangement of five and seven cylinders these are more favorable as regards transverse oscillatory strain. For example, the vibrographic measurements which were taken in the motorship *Seattle* of the Hamburg-Amerika Line, are shown in Table II. This refers to a double-acting two-stroke cycle seven-cylinder engine of 700 millimeters cylinder bore and 1200 millimeters stroke. The measurements were made on the upper cover of No. 1 cylinder and were recorded at various speeds of revolution. It will be seen that in the whole range of working ± 2/10 millimeter was measured as the maximum side displacement. These displacements were so slight that they could not be ascertained without the aid of a very sensitive recording apparatus.

(c) CONSTRUCTIONAL DETAILS.

Figure 13 shows a section through the center between two cylinders of a compressorless double-acting two-stroke cycle engine of 12,000 B.H.P. In this illustration, a through-bolt can be seen extending from the base plate to the upper edge of the working cylinder, which takes away the tension stress from the cast-iron ports and the working cylinder. The bolts, which are also available for connecting the foot of the column with the base plate, provide for adequate transverse strength at this point.

Figure 14 is a section through a working cylinder of the same type of 600 millimeters diameter and 900 millimeters piston stroke. These are compressorless engines without camshafts and without mechanically-operated valves. Special attention has been paid to the cooling water. The joint between cylinder cover and liner is placed as near as possible to the center of the cylinder, and therefore the liners are as short as possible. This is favorable inasmuch as the temperatures of the liners can be kept low, as the combustion space is only surrounded by the cylinder cover.

The results of temperature measurements are shown in Figure 15. It will be seen that the maximum temperature at an indicated pressure of 6.3 kilograms per square centimeter was only 130 degrees C. at the hottest point. By using a flat cylinder cover, *i.e.*, with the liners carried up higher, as is mostly the case, the temperature at this point would be about 325 degrees C. The lubricating condition of the liners is favorably influenced by the low temperature. For the cooling of the liners a definite control of the water is provided, which prevents, along with the low temperatures, any salting or incrustation of the surface of the liners in the cooling-water space at low temperatures. The cooling of the liners still remains satisfactory after many years of service, and there is no necessity to take these out for cleaning purposes. In case of need, however, the short liners—those on the lower side—can be removed very easily.

Figures 16 and 17 show how to dismantle the lower cylinder cover and liner. This design is extremely simple, the cover above the crankcase remaining in position. One usually hears from supporters of the single-acting engine that a disadvantage of double-acting engines is that they have lower cylinder covers and stuffing boxes for the piston rods, and that their removal is difficult. It may be said, in reply to this, that there is no necessity to remove the lower cylinder cover for the purpose of overhauling the stuffing boxes. The stuffing-box rings, which are fastened together by means of screws, are let down in the crankcase. If one further considers that the question of removal of the lower cylinder cover would only arise when these parts are to be changed, and also that the weight of the parts in question is

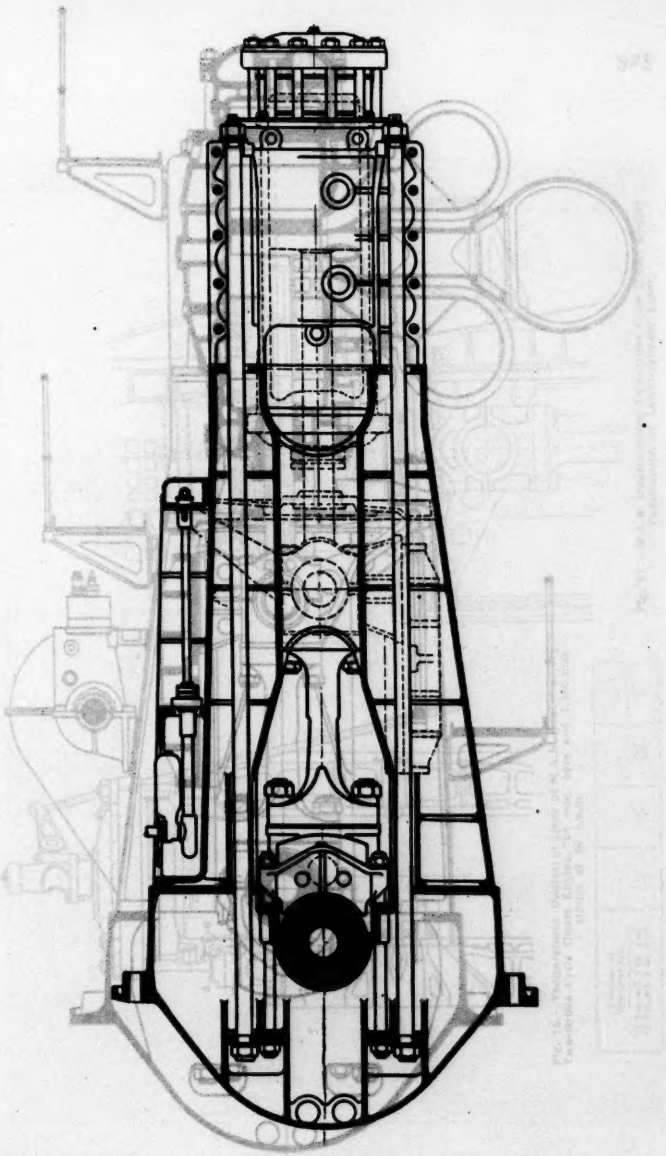


FIG. 13.—M.A.N. DOUBLE-ACTING TWO-STROKE CYCLE DIESEL ENGINE—TRANSVERSE SECTION IN WAY OF THROUGH-BOLTS.

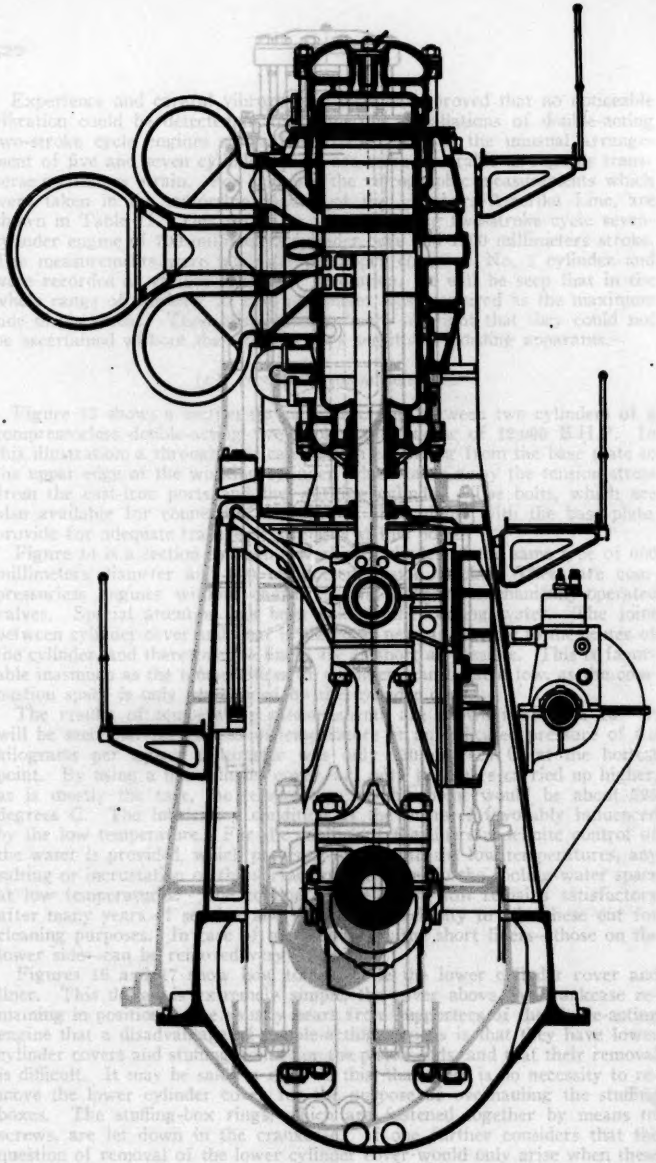


FIG. 14.—TRANSVERSE SECTION OF M.A.N. DOUBLE-ACTING TWO-STROKE CYCLE DIESEL ENGINE WITH DIRECT FUEL INJECTION (COMPRESSORLESS).

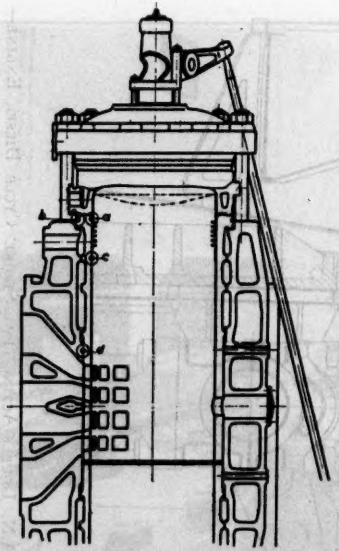


Fig. 15.—Temperatures (Celsius) at Liner of M.A.N. Double-acting Two-stroke Cycle Diesel Engine, 700 mm. bore and 1,200 mm. stroke at 90 r.p.m.

Position of Measurement.	a	b	c	d
At [illegible] kg./cm²	110°	50°	73°	50°
At [illegible] kg./cm²	115°	51°	80°	51°

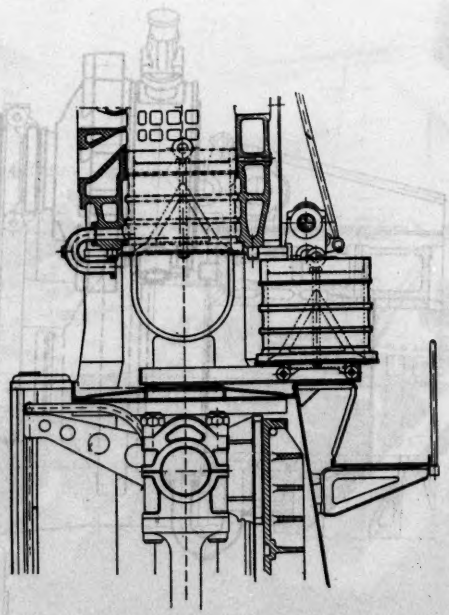


Fig. 17.—M.A.N. Double-acting Two-stroke Cycle Diesel Engine Construction of Lower-cylinder Liner.

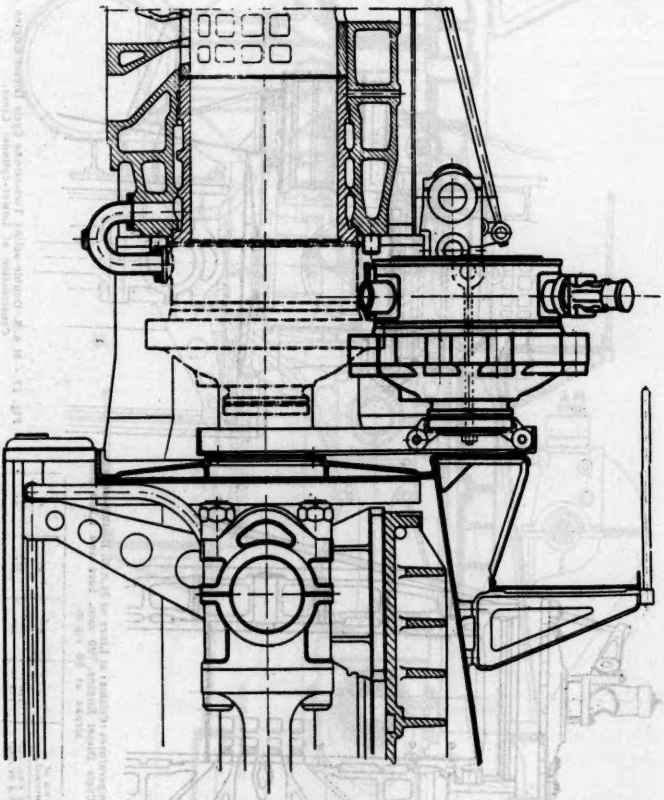


FIG. 16.—M.A.N. DOUBLE-ACTING TWO-STROKE CYCLE DIESEL ENGINE—CONSTRUCTION OF LOWER-CYLINDER COVER.

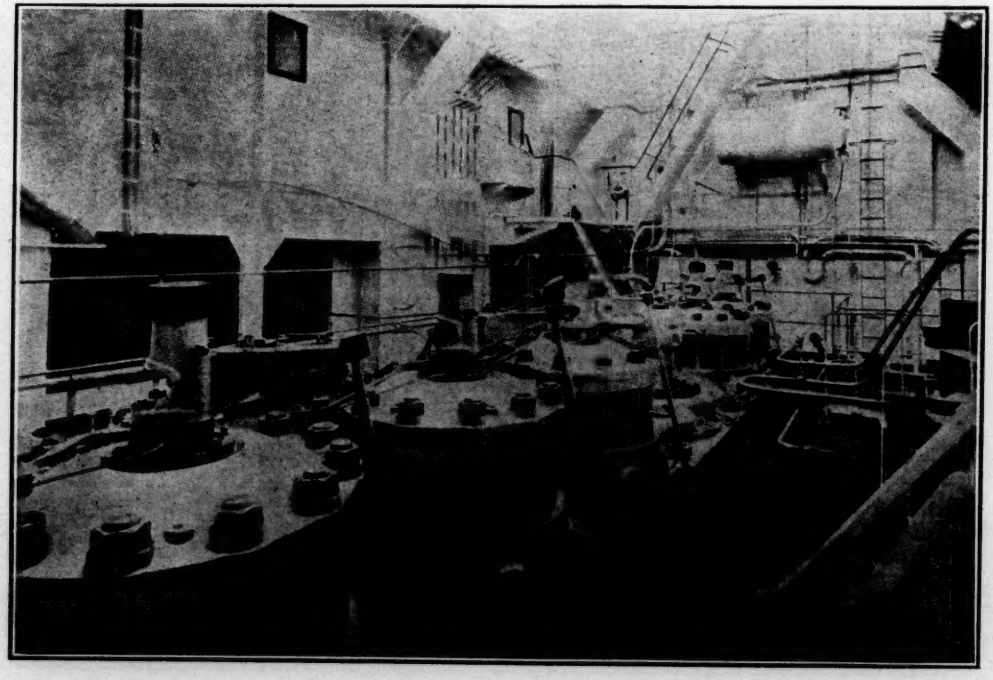


FIG. 18.—VIEW OF CYLINDER TOPS OF M.A.N. DOUBLE-ACTING TWO-STROKE CYCLE DIESEL ENGINE.

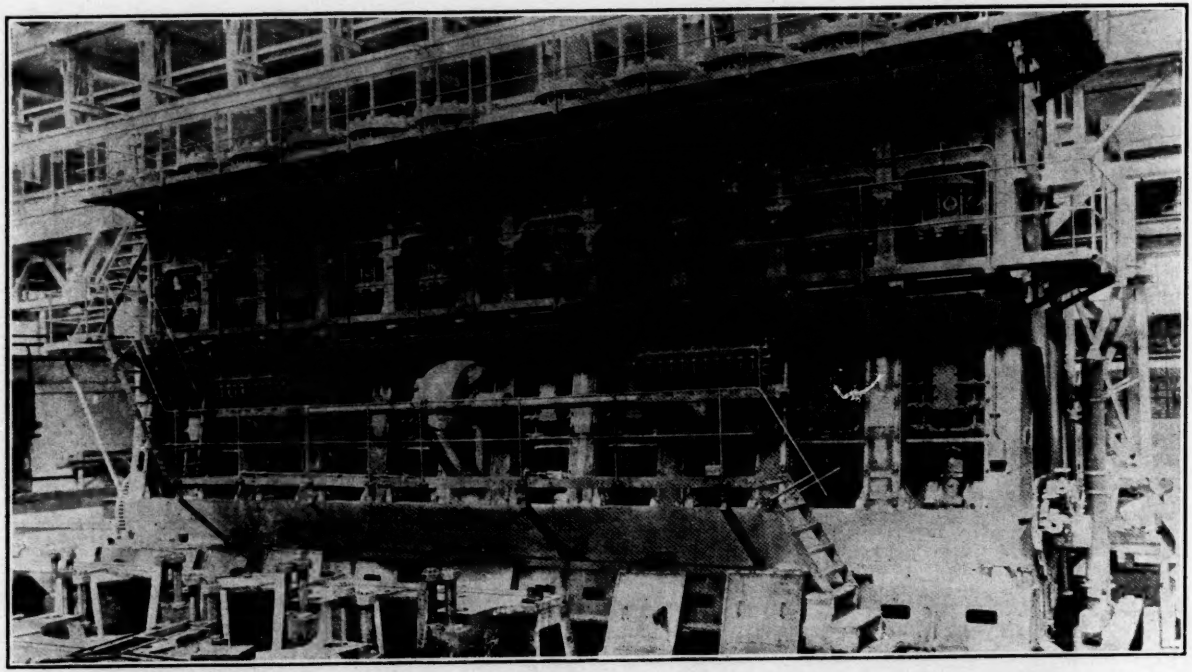


FIG. 19.—M.A.N. DOUBLE-ACTING TWO-STROKE CYCLE DIESEL ENGINE OF 11,700 B.H.P.—VIEW FROM STARTING-PLATFORM POSITION.

Fig. 19.—M.A.N. Double-Acting Two-Stroke Cycle Diesel Engine of 11,700 B.H.P.—View from Starting-Platform Position.

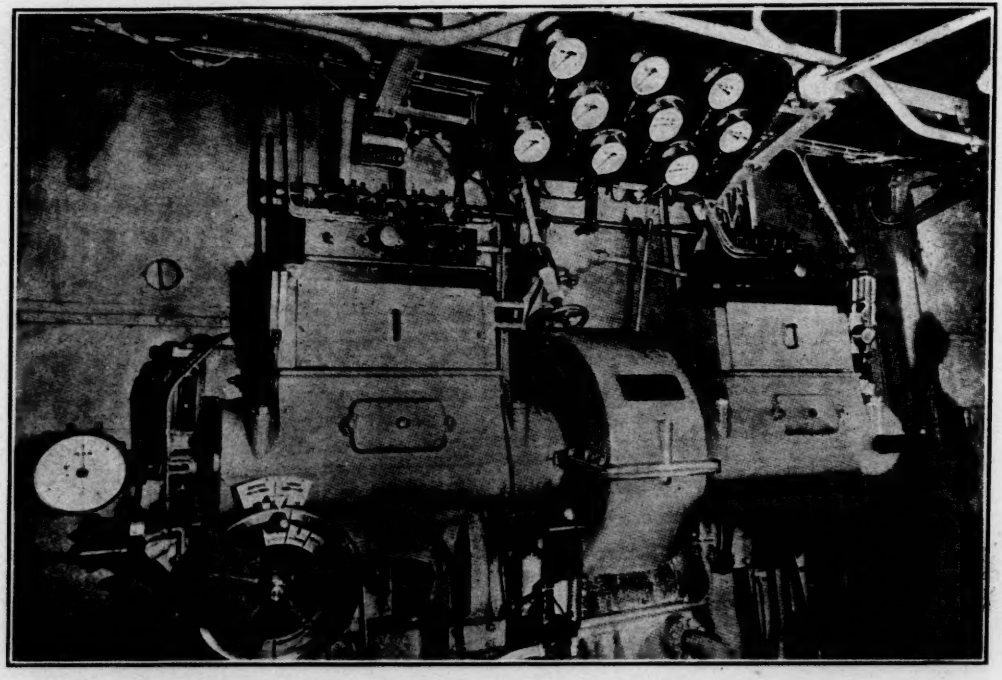


Fig. 20.—Maneuvering Stand of M.A.N. Double-Acting Two-Stroke Cycle Diesel Engine.

much less and the dimensions much smaller than those of all other types, it is obvious that the double-acting type of engine is not behind the single-acting engine as regards ease of dismantling the various parts.

The camshaft has been placed lower (see Figure 18) in order to make it easier to reach the upper cylinder cover for the purpose of overhauling the cylinder cover and piston. The design of the crossheads of the working cylinder is kept one-sided. Contrary to the usual double-sided guide arrangement, a smaller depth is necessary in this case, and likewise a smaller central distance between the cylinders. It is not without importance that the guides increase the strength of the engine columns, as can be seen from Figure 19. In connection with the working cylinders lying above the columns, the entire engine gives the most favorable conditions for strong horizontal and transverse construction.

The casings for the trombone cooling are fitted in the columns. The trombones themselves run in front of the reverse wall of the columns. The engines which have so far been placed in service have proved that no lubricating oil drips into the circulating fresh water, and it is well known that attention must be paid to the condition that lubricating oil should be kept away from the cooling water. An arrangement for separating the oil from the fresh cooling water is not necessary in the case of the M.A.N. construction.

In order to keep down the requirements of the fuel-pumping gear and to deliver the fuel to all fuel valves at the same instant referred to the opening of the fuel-valve needles, the mechanism of the individual fuel-pump pistons has been timed according to the firing order. The fuel pump is placed together with the manometers and tachometers, and arranged so as to enable a general view to be obtained. Fastened to the driving wheel of the fuel pump itself is a governor, which works the governing shaft of the fuel pump through a short rod. This prevents an excessive number of revolutions beyond about 5 per cent, and, in addition, the range of the governor can be adjusted by means of a spring. The starting and maneuvering gear (Figure 20) is connected with the engine telegraph by means of a patented M.A.N. arrangement, so that mistakes in maneuvering are avoided. For maneuvering purposes, a hand-wheel is turned in one direction over the reverse and starting position into working position. The regulation of the fuel injection and the direction of rotation of the engine are influenced by the position of the hand-wheel.

The cooling of the working cylinders and cylinder covers can be effected by sea water. The working pistons are cooled by fresh water, in order to avoid corrosion in the piston rod. Fresh-water cooling of the pistons requires, of course, a separate fresh-water cooler.

* * * * * * * *

The foregoing remarks are intended to show the advantages offered by the double-acting two-stroke cycle Diesel engine, especially for powers exceeding those of other types of engines. These advantages will doubtless have an important bearing on the future development of this system. The fundamental conditions for developing the double-acting engine as a fast-running unit are much more favorable than in the case of the single-acting two-stroke cycle type, and it appears that this will play an important part in shipping in the near future.

The M.A.N. have been working on the development of the double-acting two-stroke cycle Diesel engine, as pioneers, for a long time, and they can look back on a large number of successful installations already completed and enjoy the benefit of the experience gained in this connection. Not only

do builders consider this type of great value, as shown in the development curves previously mentioned, but this claim is confirmed by the fact that the German cruiser *Ersatz Preussen* will be fitted with M.A.N. double-acting two-stroke cycle Diesel engines with a total power of 50,000 E.H.P.—"The Shipbuilder," July, 1929.

LATEST RESULTS FROM TUNGSTEN CARBIDE TOOLS.*

DIRECTIONS FOR THE DESIGN AND FABRICATION OF TOOLS—RESULTS OBTAINED IN CUTTING CAST IRON AND ALLOY STEELS—SOME UNSUCCESSFUL APPLICATIONS.

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The brittleness of tungsten carbide materials suggests the use of a large—and especially a thick—tip, in relation to the size of the cut to be taken. The factors limiting the tip size may be the cost of the material, the necessity for ample support of the cutting tip by a stronger and tougher material, and the space available for the operation of the whole tool. It is also necessary, in designing a tool, to consider chip space, especially in the case of a multiple-edged tool, in order that there will be no likelihood of the chips becoming packed on the tool face so that chipping of the edge results.

The comparative weakness of this material demands as sharp a cutting angle as possible, to reduce the chip pressure. In order to use a large rake angle, however, without crumbling at the edge, great rigidity is essential. As more rugged machines are built, it will become possible to use steeper cutting angles. The angles depend, of course, on the material being cut, the type of tool used, and the speed, feed, and condition of the machine. As an example, lathe tools for malleable cast iron are used with a front rake of 5 degrees and a side rake of 5 degrees; these angles can probably be increased for steel if there is freedom from vibration in operation.

As heat generated in the tool is relatively unimportant, it is possible to use broader-nosed tools of tungsten carbide than of high-speed steel, thus distributing the cut over a wider area and lengthening the life of the tool.

CUTTING TOOL TIPS TO SIZE IS NOT AN EASY OPERATION.

It is usually impossible to purchase a piece of tungsten carbide alloy of the size required in a tool. Hence, the first problem encountered, if the user wishes to make his own tools, which may be economical if quantities are used, is that of removing from a small bar of the alloy a piece of the desired size and shape. The bar of alloy should be purchased with its application in mind, so that two dimensions of the proposed tip are correct. Then it would appear to be merely necessary to cut off the length desired by means of a thin grinding wheel of the composition recommended for grinding tungsten carbide.

It is not so simple, however. It has not been found possible to cut completely through even a thin piece of this alloy with a grinding wheel; the piece always fractures before the job is finished. By feeding the disk carefully so as not to heat the alloy excessively, and by turning the piece so as to grind into two opposite sides, or perhaps into all four sides, it may

* Abstract of a paper read before the meeting of the American Society of Mechanical Engineers, at Rochester, May 14, 1929.

be possible to cut away two-thirds of the section before the piece cracks. The discouraging feature is that, even with these deep notches carefully ground, the fracture is no more likely to occur between the notches than anywhere else—with the limitation, of course, that it always starts in one of the notches.

To grind on one side only (one of the broader sides if the section is not square) directly into the piece, until it cracks, is as good as any method. When this is done, the feed of the wheel into the work, within reasonable limits, appears to have little influence on the result. The notching should not be attempted by hand, but should be done on a small surface grinder. The authors have used a 7-inch wheel, not less than ⅜ inch thick, on the circumference of which has been dressed a rim approximately 1/16 inch thick and ¼ inch wide, with fillets at the shoulders and a rounded edge.

HOW TO MOUNT TIPS IN HOLDERS.

The next step is mounting the tips in the holders. A firm support is of paramount importance. The tip must fit its seat closely, and should have contact on at least three sides. Blades of multiple-edged tools, such as reamers and facers, may be made of thin flat pieces of the alloy, which should be mounted in slots sawed at angles of 10 to 12 degrees with the direction of motion at the cutting points. This method results in locking the blades against the pressure of the chip.

The brazing may be done by heating the holder and tip in a furnace having an atmosphere of nitrogen or hydrogen, using copper as the brazing material. Good results can also be obtained without such a furnace, by heating in the flame of a blowtorch, provided a brass alloy be substituted for the copper. The strength of a well brazed joint of brass may not be so great as that of an equally good copper joint; but, owing to the fact that the brass flows more freely, a complete filling of the joint can be made much more easily under the blowtorch with brass than with copper.

GRINDING TUNGSTEN CARBIDE TOOLS.

The tip or blade may be ground to its final shape after mounting. A fixed position on a surface grinder is likely to result in a rounding or crumbling of the point; consequently, it is advisable to grind by hand, moving the tool across the side of the wheel. Good results in roughing have been obtained with 60-I to 80-I Crystolon wheels. For finishing, a 100-I Crystolon wheel is satisfactory. The wheel speeds are approximately 3400 revolutions per minute. If the grinding has been properly done, it is seldom necessary to resort to lapping as the finishing operation.

USING TUNGSTEN CARBIDE TOOLS FOR CUTTING MALLEABLE CAST IRON.

One of the operations on malleable-iron castings to which these tools are applicable is the breaking down or chamfering of the flanges of differential case castings. The operation is the first performed on the rough casting. One typical case requires a chamfer of about ¼ inch on a flange 9 inches in diameter. According to the best results obtained prior to the use of tungsten carbide (with a speed of 72 revolutions per minute and a feed of 0.014 inch), the production per grind averaged 200 pieces; by simply replacing the former tool with one of tungsten carbide, the average production jumped to over 7000 pieces per grind.

On the same lathe set-up, the flange is rough-faced. With the tool formerly used, making a cut 1/16 inch deep, and a feed of 0.014 inch, at 170

feet per minute, the tool life was about 150 pieces per grind. Tungsten carbide tools are averaging 700 pieces per grind. The speed of these operations is at present limited by other tools in the set-up, notably a reamer.

The finish-facing of the same flange is performed on another lathe. High-speed steel tools formerly used produced an average of 400 pieces per grind. Tungsten carbide tools operating at a speed of 300 feet per minute, with 0.013 inch feed, average 12,000 pieces per grind. These tools do not give quite as smooth a finish on the work as high-speed steel, but the great advantage of the new tools, aside from increased production, is the elimination of size variation. It was necessary for the operator to readjust the set-up several times during the life of a steel tool; with the tungsten carbide tool, no adjustment is required throughout a day's run, and no difference in size between the first and last piece can be detected.

Another malleable cast iron application is that of line-reaming pinion bores of differential carriers. A double reamer 3⅜ inches in diameter, each section having six blades, is run on one such job, at 80 revolutions per minute, removing about 0.015 inch with a feed of 0.052 inch. High-speed steel reamers produced an average of about 1500 pieces per grind, and each reamer had a total life of about 6000 pieces. The tungsten carbide reamers average 9500 pieces per grind, and the life of a reamer is approximately 28,500 pieces. In this application, better finish as well as less size variation is obtained with tungsten carbide. The rate of production, furthermore, has been increased approximately 20 per cent, as the high-speed reamers were run at half the speed mentioned, with a little greater feed.

In these applications the use of tungsten carbide tools has progressed beyond the experimental stage. These tools have been definitely adopted as standard on these jobs.

CUTTING ALLOY STEEL WITH THE NEW TOOLS.

Everyone has been cautious in applying tungsten carbide to the machining of steel. Nevertheless, there is at present one important job on which a saving is being made by its use. Bevel drive gear forgings, usually of 3½ per cent nickel steel, were formerly made with minimum machining allowances of 1/16 inch on the back and 1/16 inch on the face. The initial operation was to take a fairly heavy facing cut from the backs of the forgings on heavy machines with high-speed steel tools. Owing to the powerful chucking required, an occasional forging was distorted so that, when it was removed from the machine after the cut, the back would not be quite flat. This condition caused trouble in subsequent operations.

The present method is to specify 0.070 inch allowance for machining on the face and to merely clean up the forge shop excess of 1/32 to 1/16 inch from the back with a tungsten carbide facing tool, on a lathe. This means that a comparatively thin cut is taken on a surface so abrasize and at a speed so high that a high-speed steel tool will not finish one piece.

The speed of the initial facing cut on a typical forging, which is from 8½ to 10 inches in diameter, is 120 revolutions per minute, or from 267 to 314 feet per minute. The feed is 0.020 inch. A tungsten carbide tool will face 150 to 175 pieces per grind and will have a total life of 3000 to 4000 pieces. It should be mentioned that some difficulty was experienced in obtaining a lathe that would stand up under the high speeds required to make this facing operation economical, and before such a lathe was used, the tungsten carbide tools gave much trouble in chipping due to lack of rigidity.

SOME UNSUCCESSFUL APPLICATIONS.

It is natural that, in seeking for applications on which tungsten carbide tools will improve the product or lower costs, some failures will occur. In the interests of all concerned, it is fully as important to record operations for which these tools seem to be unadapted as it is to broadcast their successes. The authors have been unable to make a cutting-off tool for automatic screw machines that operates successfully; every tool chips very quickly. They have not been able to make a tungsten carbide tool work satisfactorily on "Mult-Au-Matics"; the large overhang permits sufficient vibration to chip the tool. This alloy has also proved ineffective in rough-turning the outside diameter of the flange of some differential cases of malleable cast iron, owing to the irregularity of depth of cut. They have also had some difficulty in either purchasing or manufacturing multiple-edged facing tools from which the tungsten carbide blades will not pull out during operation.

In conclusion, it is evident that tungsten carbide alloys are to have an important place in the machining of iron and steel. The authors believe that the future of these alloys as factors in efficient manufacturing is in the hands of neither the makers of the tool materials nor the builders of the machines, but the users of machine tools.—"Machinery," June, 1929.

BOOK REVIEW.

JAHRBUCH DER BRENNKRAFTTECHNISCHEN GESELLSCHAFT (JOURNAL OF COMBUSTION ENGINEERING SOCIETY). Volume 9, 1928. Published by Wilhelm Knapp, Halle (Saale), 68 pp., price 7.50 R. M.

To those reading technical German and interested in development of power from combustion the yearly appearance of this booklet dealing with the advances in this important art is welcome. The following papers are presented in this year's volume:

(1) High pressure steam and the Diesel engine by F. Flatt.
(2) The chemical utilization of coke oven gases by J. Bronn.
(3) Nature and forms of combustion by Prof. Aufhaüser.
(4) American Investigations of Exhaust Gases of Automotive Vehicles by W. Liesegang.

The first paper is written by an advocate of high pressure steam and attempts to prove that the solution of the power problem on land and sea, for stationary and locomotive use lies in the use of coal, preferably in powdered form, and high steam pressures. The interesting discussion by well known engineers, which followed, saved the Diesel engine from complete annihilation.

The second paper deals with a subject very important to Germany which has neither oil nor natural gas and therefore cannot afford to waste or even misuse the by-products available in coke oven gases, and describes means for separating gaseous components through low temperature cooling. Uses developed for such gases are also described.

The third paper deals with the fundamentals of combustion and the author comes to the conclusion that time and not temperature is important in ignition problems, and that the problem

of "combustion velocity" is by far the most outstanding for the future development of all forms of combustion.

The fourth paper is a review of the work done in 1927 by the U. S. Bureau of Mines to assist in the designing of the ventilation system of the Holland Vehicular Tunnel between New York and Jersey City, N. J.

E. C. MAGDEBURGER.

ASSOCIATION NOTES.

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